WEAR AND METALLIC CONTACT STUDIES OF EN 31 STEEL USING ELEMENTAL SULFUR IN WHITE OIL

A THESIS

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CERTIFICATE

Certified that the work presented in this Thesis has been carried out by Mr.V.P. Chawala, Metallurgical Engineering Department, under our joint supervision and has not been submitted elsewhere for a degree.

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CHAPTER - ONE LUBRICATION, FRICTION, AND WEAR FUNDAMENTALS.

1.1 LUBRICATION AND LUBRICATION REGIMES(1)

Lubrication is an essential feature of all modern machinery. Lubricants are interposed between surfaces and components in relative motion. Relative motion of surfaces results in friction and wear of components and the science of lubrication grew out of the need for reducing both friction and wear. The detrimental effects of friction are:-

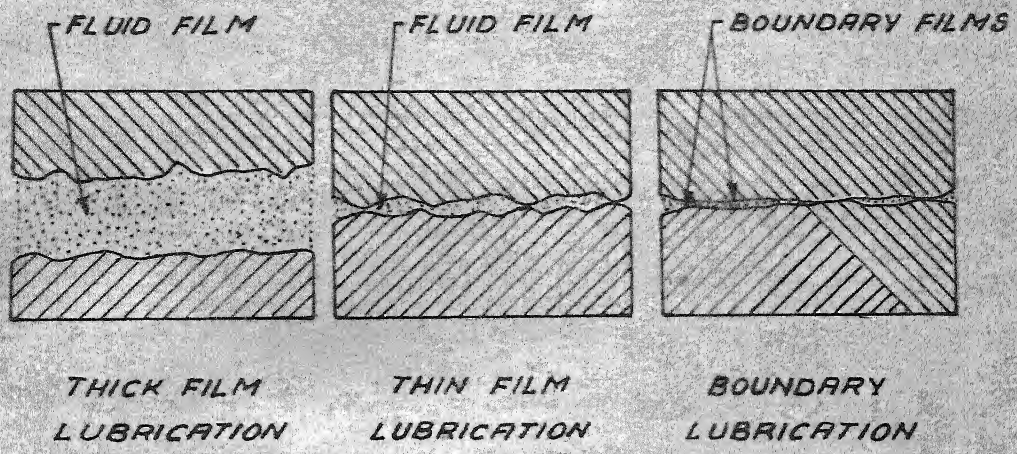
* The resisting force opposing the motion of bodies.
* Loss of power owing to the work done against friction.
* Temperature rise and subsequent surface damage.

Lubrication may be envisaged as a situation where two surfaces are partially or completely separated by lubricant film. The extent to which a surface is separated from its mating surface depends upon the lubrication regime. Lubrication regimes can be broadly distinguished by two main types:-

(A) Thick Film Lubrication Regime.

(B) Thin Film Lubrication Regime.

Thick film lubrication occurs when mean film thickness of the lubricating film is large enough compared to the



SCHEMATIC ILLUSTRATION OF RELATION OF SURFACE ROUGHNESS TO FILM THICKNESS UNDER CONDITIONS OF THICK FILM, THIN FILM, AND BOUNDARY LUBRICATION

FIG.- 1-A

height of asperities (rough spots of a surface since all surfaces are rough on a microscopic scale). Thick film lubrication tends to occur at high speeds and moderate loads, and in this type of lubrication, lubricant viscosity is the most important factor.

Thin film lubrication occurs when the mean film thickness of the lubricating film is thin i.e. of the same order of magnitude as the height of the asperities. Thin film lubrication tends to occur at low speeds and high loads. Most of the load is supported by the asperities in contact in this case and metal-lubricant interaction becomes important in the case of thin film lubrication. This type of lubrication occurs for example in metal-cutting, and between piston-ring and cylinder.

1.2 THICK-FILM LUBRICATION

Complete separation of surfaces is possible in Thick Film Lubrication. The regime of thick film lubrication can be further classified into three main types as follows[2].

(1) Hydrodynamic Lubrication

(2) Elastohydrodynamic Lubrication

(3) Hydrostatic Lubrication

<u>Hydrodynamic Lubrication</u>:- It occurs when the sliding surfaces are completely separated by a layer of lubricant film. In this case, a fluid pressure is generated by the motion of the bearing surfaces. The pressure supports the load, maintaining a film thick enough to prevent metal to metal contact.

The thickness of the lubricating film, in hydrodynamic lubrication, is an approximate function of $\left(\frac{\mu N}{P}\right)$ parameter,

where, μ = Viscosity

N = Rotational Speed of Journal

P = Pressure

The lubrication is governed by the properties of the lubricant, in particular, its viscosity. The only resistance

to the relative movement of the surfaces is the force required to shear the lubricant. If the values of $\left(\frac{\mu N}{P}\right)$ are relatively high, hydrodynamic lubrication will prevail when the shaft will be supported by a wedge of lubricant and coefficient of friction will depend upon the viscosity of the lubricant.

<u>Elasto-hydrodynamic Lubrication</u>:- This may be considered as a special case of hydrodynamic lubrication where the elastic deformation of the surfaces and pressure effects on viscosity become important. In simple hydrodynamic lubrication, we may obtain the film thickness and friction coefficient by the following typical relations:

$$h \propto \frac{\mu u^a}{w} \quad \text{and} \quad f \propto \frac{\mu u^b}{w}$$

where,

μ = viscosity of the fluid

u = sliding speed

w = normal load

h = minimum film thickness

f = friction coefficient

a, b = constants that depend on bearing geometry.

In the case of elastohydrodynamic lubrication film thickness equations are developed on the basis of somewhat complex analysis of the hertzian deformation coupled with

pressure effects on viscosity. Typical film thickness equation for two cylinders in rolling/sliding contact can be expressed as follows:

$$\frac{h_o}{R} = 1.19\left(\frac{u\ n_o \alpha}{R}\right)^{0.73} \left(\frac{E_L\ R}{w}\right)^{0.11}$$

h_o = Minimum Film Thickness

E_L = Reduced Elastic Modulus

$u = \dfrac{u_1 + u_2}{2}$ Where u_1 , u_2 are peripheral velocities of the two cylinders.

α = Pressure Coefficient of viscosity

w = Load per unit width

R = Reduced Radius

n_o = Inlet Viscosity

This equation may be compared with the film-thickness-equation for hydrodynamic lubrication. The film thickness in EHL varies very little with load unlike hydrodynamic film thickness. Also, as expected the film thickness is significantly influenced by α , the pressure coefficient of viscosity.

The main aspect of EHL theory is that net film thickness will be significantly higher than when calculated on the basis of hydrodynamic theory alone. Thus in many

concentrated contacts, for example gears, the film thickness is sufficient to prevent metal to metal contact.

Hydrostatic Lubrication:- Hydrostatic lubrication is accomplished by introducing the fluid from outside with a pressure sufficient to separate the surfaces and balance the load even when there is no relative motion. Bearings that depend on hydrostatic lubrication are called "Externally Pressurised". The hydrostatic principle is specially useful when starting under load and is often relied upon for that purpose in bearings which are otherwise "Self-Acting". The lubricant is supplied at a high pressure to a pocket in the bearing which lifts the shaft and the main advantage of the scheme is that surfaces can be separated by full fluid film even at zero speed and thereby the wear at low speeds is minimized.

1.3 THIN FILM LUBRICATION

In many contacts the operating conditions are such that hydrodynamic or EHD films cannot exist. The lubrication in such cases has to be of the thin film type where films of molecular dimensions are responsible for lubrication. This situation is called boundary lubrication. The properties of the films are no longer the same as in the bulk form. In such situations, there is partial contact through such films and the effectiveness of the film is related to the extent they prevent contact through such films.

Boundary lubrication tends to occur at higher loads, lower speeds, and increased surface roughness.

According to Hardy's Theory[1], the boundary lubrication is due to the polar group in the molecules adsorbed on friction surface. It is assumed that the adsorbed molecules would orient themselves with polar groups adhering to the metal. The contact between two boundary lubricated surfaces would take place between the non polar groups whereby slip would be permitted, friction reduced, and the metal protected. This is explained in detail as follows.

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In boundary lubrication, ordinary mineral oils are not effective lubricants as these molecules cannot tenaciously 'stick' to the surface. This is because these molecules are weakly adsorbed on the surface. If in the mineral oil small quantities of 'polar' compounds are introduced, they get preferentially and tenaciously adsorbed on the surfaces. These films offer good protection as they significantly reduce metal contact through films. These boundary films are only one or two molecular layers thick. The polar compounds normally used are those with a long 'backbone' of carbon atoms with an active polar end group. The normally used compounds are alcohols, amines, and acids which contain active OH, NH_2, and COOH groups respectively. The polar end groups are attached to the surface and there is strong lateral attraction between the chains. If two surfaces covered with such films come into contact they tend to slide over their outermost faces. Some penetration may occur but it is far too less than would occur if only a mineral oil is used. This view of boundary lubrication is now commonly accepted mainly due to the extensive investigations of Bowden and Tabor[2].

Detailed studies by Bowden and Tabor[2] have shown that boundary films are effective only till the melting point of the films. In some cases, the films may involve soap formation and effective lubrication extends upto the melting point of the soap. This temperature would be higher than the melting point of polar compound itself.

The maximum temperature to which polar compounds are effective is of the order of 200°C. When severity of condition is such that higher surface temperatures are involved, protection is obtained by using the so called EP (Extreme Pressure) additives.

1.4 MIXED FILM LUBRICATION[1]

The regime where hydrodynamic and boundary effects co-exist is called mixed lubrication regime. This means that a part of the load is supported by solid contact (Boundary or semidry lubrication) and the remainder is supported by fluid film. As the film thickness increases, a progressively smaller fraction of the surface incurs the high friction of boundary contact. Owing to the simultaneous action of boundary and hydrodynamic films, the friction condition is called 'mixed friction'.

1.5 FRICTION FUNDAMENTALS

As early as seventeenth century, Amontons proposed the following laws of friction.

(a) The friction force was proportional to normal load.

(b) The friction was independent of the apparent area of contact.

In friction, nature of the surfaces, physical and chemical constitution of base metals, and interaction of these surfaces in contact are important factors. With vast amount of knowledge in basic sciences available today, it is still difficult to formulate a universal friction theory. However, the main theories are as follows.

(A) Molecular Attraction Theory

(B) Molecular Mechanical Theory

(C) Adhesion Theory

Molecular Attraction Theory:- Coulomb [3] considered that some kind of molecular attraction might be responsible for friction force. Tomilson [4] was convinced that deformation placed the surfaces at a distance that compared favourably with atomic distances and crystal planes and he ascribed friction force due to atomic interaction. Electrostatic attraction theories and electromagnetic theories have been proposed but little confirmatory work has been done to allow any comment.

Molecular Mechanical Theory:- This theory comes from Russian School mainly from Kraghelsky[5]. Kraghelsky attributes friction to the formation of a deformation wave that travels ahead and along the sides of the indentor; This he feels is produced as a result of intermolecular forces. The wave seems to proceed by material being lifted upto the height of the crest. Adhesion will increase this height, and the yield point of the metal determines the form that the crest will take.

When the deformed material is elastic, the hysterisis losses are the determining factor, whereas for plastic materials, friction coefficient will depend on the ratio between plastic deformation in the tangential and normal direction.

Adhesion Theory:- This theory is due to Bowden, Moore, and Tabor[6]. According to this theory, both plastic flow and adhesion occur due to friction and Bowden et al. proposed a two term formula for frictional force, F.

$$F = A_1P_1 + A_2P_2$$

The first term is due to displacement of plastically deformed metal and the second is due to adhesion. A_1 is the cross sectional frontal area of the slider and P_1 is the resistance to plastic displacement. A_2 is the area over which metallic junctions are formed and P_2 is the strength of these junctions.

1.6 WEAR AND TYPES OF WEAR(3,8)

Wear is a surface phenomena which is defined as the loss of material during the rubbing of the surfaces due to mechanical action of sliding or rolling forces. Wear may also be due to corrosion but mechanical force is involved in this case also which removes the metal as corrosion product.

All mechanical components that undergo sliding or rolling contact are subject to some degree of wear. Typical of such components are bearings, gears, seals, guides, piston rings, splines, brakes and clutches. Wear of these may range from mild polishing type of attrition to rapid and severe removal of material with accompanying surface roughening.

Types of Wear:- Wear can be of several types. One of these types may predominate in a given situation, or several types may be operative at the same time. Also, one type of wear may initiate a second type of wear process. For example, metallic wear can generate hard particles and can therefore lead to abrasive wear. Also, abrasive wear can quickly destroy the lubricating film and can lead to galling type of wear.

For many years, there has been considerable disagreement regarding the forms or types of wear. The terminology of wear was unsettled, and basic definitions were not standardised. None of these problems have been completely resolved though areas of general agreement continue to emerge.

Burwell and Strang[7] have classified wear into five primary types as follows:

1. Adhesive Wear
2. Abrasive Wear
3. Erosive Wear
4. Corrosive Wear
5. Surface Fatigue Wear

In addition, there are other types of wear which, although not regarded as primary, are afforded separate status as follows[8].

1. Erosion-Corrosion
2. Fretting
3. Cavitation-Erosion

Explanation of above types of wear follow in the following text.

Adhesive Wear:- This wear occurs when two metallic surfaces slide against each other under pressure. Asperities bond at the sliding interface under very high local pressures. Subsequent sliding forces fracture the bonds, tearing material from one surface and transferring it to the other. This results in the formation of minute cavities on one surface and minute projections on the other - which in turn can lead to further damage. The process may also result in the formation of loose wear particles, and these are work hardened hard particles and therefore lead to abrasive wear.

Abrasive Wear:- Abrasive wear is displacement of material from a surface by contact with hard asperities of a mating surface, or by contact of hard abrasive particles that are moving relative to the wearing surface. When hard particles are involved, they may be trapped between two sliding surfaces and abrade one or both of them, or they may be embedded in either of the surfaces and abrade the opposing surface. Abrasive wear can occur in dry state as well as in lubricated state.

Erosive Wear:- Erosive wear is abrasive wear involving loss of surface material by contact with a fluid that contains particles. Relative motion between the surface and the fluid is essential to this process and the force on the particles that actually inflict the damage is applied

kienetically. Erosive wear can be,

* Liquid impingement erosion

or

* Abrasive erosion and impingement erosion

Liquid impingement erosion is caused by liquid droplets carried in a rapidly moving stream of fluid.

Abrasive erosion is caused when the relative motion of suspended particles in a fluid is nearly parallel to the eroded surface.

Impingement erosion is caused when the relative motion of suspended particles in a fluid is nearly normal to the eroded surface.

Corrosive Wear:- Corrosive wear is a type of mechanical wear in which chemical or electrochemical reaction with the environment significantly contributes to the wear rate.

In some cases, chemical reaction takes place first and is followed by the removal of corrosion products by mechanical action. However, mechanical action may precede chemical action and may result in the formation of very small particles of debris, which subsequently react with the environment.

Surface Fatigue Wear:- This is a special type of surface damage whereby particles of metal are detached from a

surface under cyclic contact stresses, causing pitting or spalling. The most impoftant surface fatigue phenomena is the Rolling Contact Fatigue.

Rolling Contact Fatigue is the result of cyclic stresses developed at or near bearing contact surfaces during operation. These stresses result in progressive deterioration of the material by one or more cumulative damage mechanisms that eventually cause initiation and propagation of fatigue cracks. In some cases, the initial stages are characterised by polished contact surfaces in which small pits are often observed. Such surface damage, if allowed to continue, also can lead to spalling in which metal fragments break free from the components, cleaving cavities in the contact surfaces.

Erosion-Corrosion:- This is a type of wear in which there is relative movement between a surface and a corrosive fluid which also may carry abrasive particles, the wear rate being directly related to the rate of relative movement. Special forms of erosion-corrosion are,

1. Cavitation erosion

and

2. Fretting.

Cavitation erosion:- This can occur on a surface in contact with a liquid that does not contain particles. Here, repeated formation and collapse of fluid-bubbles at the surface

imposes large repetitive contact stresses that cause pitting or spalting.

Fretting:- This is sometimes known as wear-oxidation, friction-oxidation, or chafing. This occur between two contacting surfaces subjected to repeated, small-amplitude relative sliding, such as from vibration, in the presence of oxygen. The damage may appear as pits or grooves, with surrounding corrosion products (oxides), on one or both surfaces.

Fretting is complex process and often involves a combination of Corrosive, Adhesive and Abrasive wear. As a result of vibration, surface-fatigue-wear may also be associated with fretting.

Kislik[9] gave more fundamental classification of wear, as follows:

(1) Mechanical Destruction of Interlocking Asperities.

(2) Asperity Fatigue.

(3) Failure due to Working.

(4) Flaking of oxides - films.

(5) Molecular Interactions.

(6) Mechanical destruction due to High Temperatures.

Classification based on wear mechanism:-[10] Wear is a process of particle-removal from surfaces rather than atomic attrition. Therefore wear can also be classified according to various mechanisms by which particles can be removed away from the rubbing surfaces. Particles can be removed from either dry surfaces or lubricated surfaces as follows:

(1) Adhesion and shear of Junctions.

(2) Surface Fatigue.

(3) Fatigue

(4) Cutting

(5) Melting

(6) Surface Reactions and Removal of loose reaction Products.

(7) Plastic Deformation and Tearing.

Above classifications are discussed in full detail by Peterson[3].

1.7 DRY AND LUBRICATED WEAR AND WEAR Vs TIME BEHAVIOUR (3)

In this section, following will be discussed as it is intimately concerned with present research work:-

A - Dry Wear

B - Lubricated Wear

C - Wear Vs Time in Lubricated Surfaces

D - Metal to Metal Contact Wear

E - Film Wear, Mechanism, and Types of films

F - Fluid Film Situation, Mechanism and Regime of Fluid Film Lubrication.

G - Transitions in Wear Vs Time Curve.

A. Dry Wear or Non-Lubricated Wear:- Metal adhesion and cold welding characterise the process of wear in the absence of a lubricant. The conditions of dry wear are difficult to define because, in most practical situations, there is some kind of "lubricant" on any sliding or rolling surfaces. In addition to the naturally occuring oxide on most metals, the atmosphere and its industrial contaminants provide a wide variety of adsorbing organic and inorganic molecules. These surface contaminants protect contacting surfaces in much the same way as boundry lubricants do, in that they prevent intimate contact between chemically active surfaces.

Only when metal surfaces are kept in an ultra high vacuum and are cleaned by an electron beam are they "truly nonlubricated". Under these conditions cold welding of the surfaces can take place immediately upon contact.

Contaminating films on metal surfaces can be penetrated under high contact stresses, resulting in cold welding of asperity contacts. If the asperity junction is stronger than the weaker of the two metals in contact, sliding motion will cause sub-surface shear of the junction and a particle larger than the junction will be torn out of the surface. It is also possible that the junction will not shear off but will grow by subsurface shear until a critical size is reached and the heavily worked junction breaks off. This process is known as "Prow Formation" and is found most often under point contact conditions involving a hard metal sliding on a soft metal.

B. Lubricated Wear:- It is difficult to form a single and universally accepted theory of lubricated wear. The reason is that there are many types of wear, many types of materials and lubricants, and many types of lubrication mechanisms. But atleast two types of wear are involved in any lubricated sliding experiments. Firstly, surface reactions are inevitable and secondly, sliding process itself modifies the lubricating characteristics of the lubricant.

C. Wear Vs Time in Lubricated Surfaces:- A variety of behaviours Fig. 1 – B have been observed on performing experiments with pin- on-disc machine (3) as follows:-

C - 1 Metal to Metal Contact Wear.

C - 2 Film Wear

C - 3 Film Wear Transitions

C - 4 Fluid Film Lubrication

C - 5 Fluid Film Transitions

C - 6 Recovery

(C - 1) Metal to Metal Contact Wear:- With a poor boundry lubricant or during "running-in" appreciable metal contact or adhesive wear will result until the surfaces become conforming.

With the poor boundry lubricant this type of wear continues throughout the experiment. If surface lubricating films are formed rapidly enough, a transition to 'Film Wear' will result.

(C - 2) Film Wear:-Some (as yet undefined) solid film is being formed on the surface and is being removed by the sliding action.

Wear will be related to the rate of film formation or the rate of removal depending upon the removal mechanism.

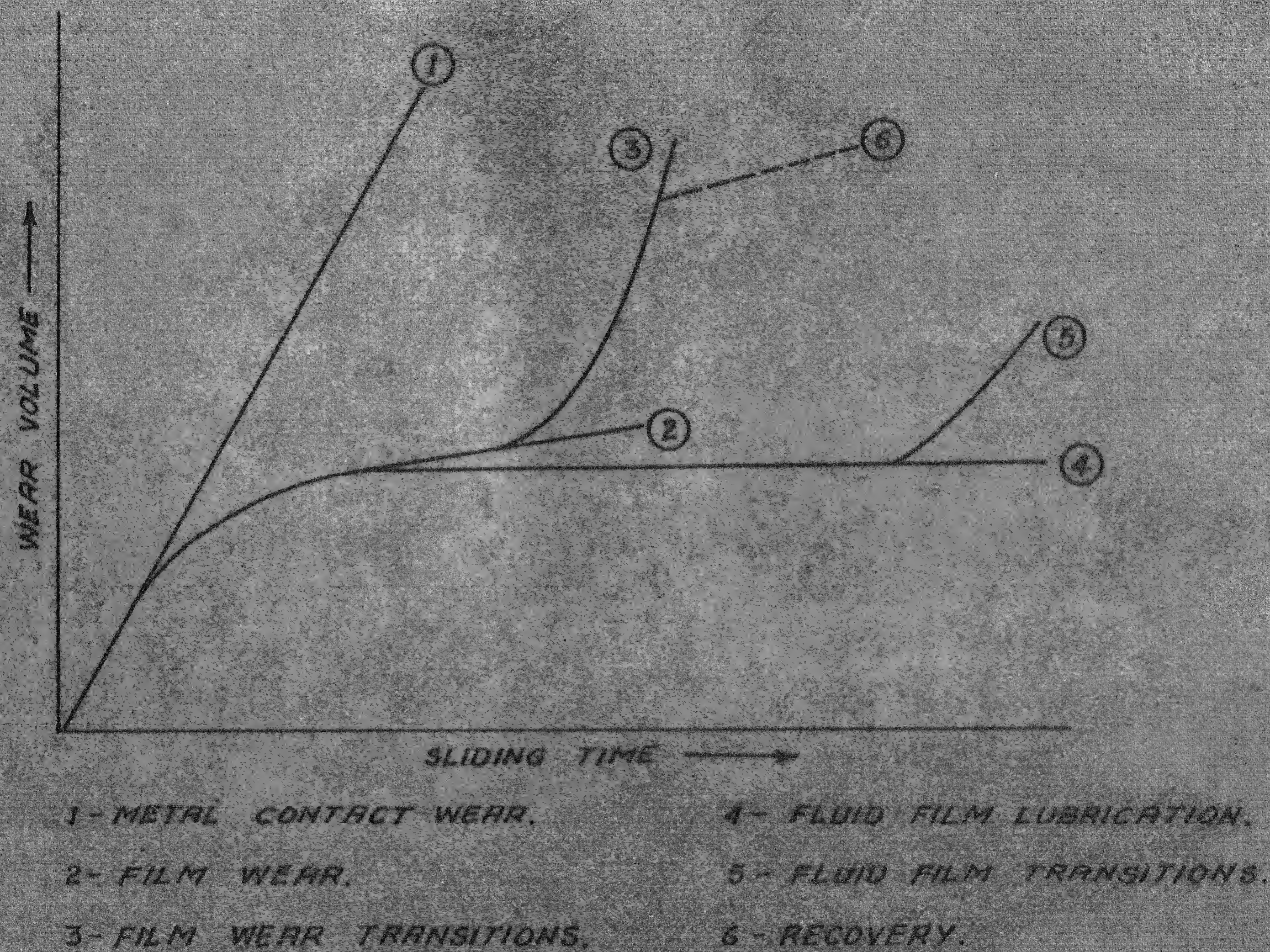

1
2
3
4
5
6
WEAR VOLUME
SLIDING TIME
1 - METAL CONTACT WEAR.
2 - FILM WEAR.
3 - FILM WEAR TRANSITIONS.
4 - FLUID FILM LUBRICATION.
5 - FLUID FILM TRANSITIONS.
6 - RECOVERY.

FIG.- 1-8

(C - 3) Film Wear Transitions:- For a variety of reasons, the initial wear rate is not maintained but increases appreciably. Some of the reasons are as follows: increased metallic contact due to increased temperature, change in the wear process, fatigue of surface, abrasion due to changes in metal and abrasion by loose debris.

(C - 4) Fluid Film Lubrication:- Wear and metallic contact ceases if the correct geometry for fluid film lubrication is established during the 'running-in' and 'film-wear' stages. This can happen due to the following changes: increase in area, change in surface contour and filling of surface irregularities with film debris.

(C - 5) Fluid Film Transitions:- Fluid film lubrication usually persists once established. However, there are mechanisms by which metallic contact can be re-established: gradual temperature build up, increased solid film thickness and shifts in relative position of specimens and dirt.
Of course, increase in load or reduction in speed can re-establish contact.

(C - 6) Recovery:- Recovery refers to the re-establishing of lubrication after one of the transitions. This is not predominant.

D. <u>Metal to Metal Contact Wear</u>:- In this concept of lubricated wear, the lubricants are not completely effective, and considerable metal to metal contact results. One usually considers that in the real area of contact, a certain fraction of the area is made up of metal contacts and the remaining area through the lubricant film. The lubricant contacts add little or nothing to wear and friction. The lubricant acts merely to reduce the amount of metallic contact. If this is so, then wear behaviour should be similar to that found for unlubricated contacts or so called dry wear. Experimentally it is found that wear volume is proportional to load and sliding distance i.e.,

$$V = C_1 . d . w$$

V = Wear Volume

d = sliding distance

w = load

C_1 = Constant

Wear volume is defined as volume of metal removed during a test and sliding distance as the distance slid for a given period of test under given set of operating conditions.

The physical basis of above equation is as follows. The fact that wear increases linearly with sliding distance or time means a steady state condition has been reached; the overall wear process is not random even if it consists of a large number of random events such as particle removal at the asperity level.

That the wear volume is proportional to load can be explained easily due to the fact that real area of contact increases with increase in load. This results in increase in metal to metal contact and hence wear.

Combined effect of time and load on wear volume is expressed by Archard's Equation (11), i.e.

$$\frac{V}{d} = \alpha \frac{KW}{3P}$$

$$(\alpha < 1)$$

V = Wear Volume

d = sliding distance

W = Load

P = Asperity yield pressure

E. Film Wear:- With certain lubricants solid films can be formed that greatly limit the amount of metallic contact; under these conditions wear of the solid film predominates. It should be noted that it is unlikely that all metallic contact can be avoided by such films.

(E - 1) Mechanism of Film Wear:- The initial wear rate is high due to the increased metallic contact associated with rough surface. At a certain point the conditions at the interface are changed sufficiently so that film formation predominates over metallic contact and generation of new surface.

Friction and wear are gradually reduced as the film behaviour becomes more and more dominant. A steady state conditions is then reached where there is balance between the film formation and film removal. Wear results in this case because sliding elements are themselves involved in the formation of film.

(E - 2) Types of Films:- There can be three types of films as follow :-

1. The "adsorbed films" that lubricate by reducing the metallic contact.
2. The soft "metal-organic reaction films" that lubricate by shearing.
3. The hard "metal-inorganic reaction films" such as oxides and sulfides that act as if they replaced the original surface with a more wear-resistant material. Some inorganic films such as chlorides can be soft and can lubricate effectively.

(E - 3) Limitations:- Above understanding of the nature of the films has been obtained from studies in which a particular film is placed on the surface and evaluated. This is a very special situation that may not be duplicated with a given compounded lubricant. What actually forms and lubricates is another matter that has not yet been known fully particularly when long wear times are considered.

Secondly, there is evidence that at least two or possibly three of these films operate together in a given situation. For example, the surface might be covered with an oxide film. On top of this, a soft reaction film or lubricant decomposition film may form. Adsorbed molecules may then occupy the outermost layers of the surface which in turn is covered by the bulk lubricant itself. Which of these most influences the lubrication under particular conditions of load, velocity, temperature and geometry is not clearly understood.

F. Fluid Film Situation:- Under certain conditions the film wear rate is not sustained; rather a decreasing rate with time is found which eventually becomes zero. This stage is named as 'fluid film wear' stage.

(F - 1) Mechanism of fluid film wear:- Various mechanism are proposed and which one operates depends upon specific system and its conditions.

(a) Load supporting fluid film:- The effect can be attributed to the formation of a load supporting fluid film at the interface and this may lead to quasi-hydrodynamic lubrication regime. It is thought that lubricant additives act as a chemical polishing agent and the load is thus distributed uniformly.

(b) Critical Stress Theory:- It is found that wear increases with time in a pin-and-disc machine until a certain pressure is reached. The wear scar increases with time but levels off to a constant value which is related to the load or stress which the contact area can withstand without further wear. This behaviour is attributed to fluid film effect. If the load is increased further the effective load on the contact is equal to the difference between the applied load W and that load which the wear scar would support without further wear Wr, wear is then given by equation[(3)] -

$$V = K\,(W - Wr)^{t}$$

V = Wear Volume

K = Constant

t = time

(c) Filling of surface roughness:- A number of investigators have found that the metal-lubricant reaction product is forced into the depressions between the asperities thus filling up the surface roughness. This produces the

equivalent of a lapped surface and thereby increasing the effective contact area and reducing the pressure.

(F - 2) Regime of Fluid Film Lubrication:- In case of fluid films, it is to be suggested that films make fluid film lubrication possible at much lower loads and moderate speeds. Also, only certain films are able to accomplish fluid film lubrication, those which possess some degree of surface adhesion. The result may be a large effective area or it may be lubricant trapping which makes elastohydrodynamic lubrication possible.

(G) Transitions:- It is so far considered that wear rate decreases with fluid film effect and wear rate remains constant with film wear effect. However, it has been noted that wear can also increase with time and is named as Transition. The reasons can be mainly,

(G - 1) Abrasion (G - 2) Surface Fatigue

(G - 1) Abrasion:- It has been found[12] with the wear of white oil that after "run-in" a constant wear rate is established for several minutes. Thereafter the wear begins to increase and this is seen at high pressures of the order of 155,000 psi. But the transition is not detected at low pressures, say 47000 psi or less. The increased wear is attributed to the transfer of particles that are harder than the parent metal. As the transfer particles

increase in size with time, the effect is greater wear of the metal by abrasion.

(G - 2) Surface Fatigue:- The increase in wear similar to white oil has been observed in cetane[13]. This transition from constant wear to high wear has been found after ten minutes of operation at low stress of 600 psi. The increased wear is suggested to be due to the fatigue of the surface by repeated stress cycles.

Increase in the wear rate can also be due to scuffing and increased temperatures at higher velocities.

1.8 EXTREME PRESSURE AND ANTIWEAR ADDITIVES (14)

There are few petroleum products currently sold which do not contain at least one additive component, and the vast majority of products contain more than one additive. The additives can be of many different types e.g., viscosity index improvers, Polymeric Dispersants, Metal Detergents, Corrosion Inhibitors, Antifoaming Additives and Load Carrying Additives. Extreme Pressure and Antiwear Additives belong to the last class i.e. Load Carrying Additives. The Load Carrying Additives are called by a wide range of names such as Antiseizure Additives, Antifriction Additives, Lubricity Additives, Oiliness Additives and Boundary Lubrication Additives.

The terms Antiwear and Extreme pressure define the conditions to which a lubricant is stressed in high load environments.

(A) Antiwear Additives:- The Antiwear Additive is normally most effective under Mixed Lubrication Conditions. Under these conditions of 'moderately loaded sliding contacts', it is generally accepted that an oil film exists between the surfaces but intermittent penetration of this film by surface asperities does occur. The Antiwear Additive probably functions by reacting with the metal asperities to

form wear resistant films. These films can help in surface smoothening thus providing hydrodynamic effects.

Antiwear Additives are employed in extensive range of lubricants, for example, automotive crank-case oils for gasoline and diesel engines, automatic transmission fluids, hydraulic fluids, Turbine oils, gear box lubricants, and aviation lubricants.

The common types of Antiwear Additives are those containing phosphorus and sulfur. Examples of phosphorus containing antiwear additives are:

Tricresyl Phosphate, Dialkyl Phosphite etc.

Examples of sulfur containing antiwear additives are:

Diphenyldisulfide

Dibenzylmonosulfide.

(B) <u>Extremepressure Additives:-</u> As the load is increased in any sliding contact, the temperature of the contact increases until, at a certain value, the bulk oil film collapses. A catastrophic increase in wear, accompanied by rapid increase in temperature, occurs leading to welding of the surfaces in the absence of an extreme pressure additive. However, when present, the Extreme pressure additive reacts with the metal surfaces to form an inorganic surface coating which can prevent welding of the surfaces and halt the catastrophic wear process.

It is probably reasonable to assume that the major difference between the antiwear and extreme pressure regions of lubrication is in the temperatures reached in sliding contact. This explains that the difference in Antiwear and Extreme pressure Additives is due to the different lubrication domains in which they become effective.

Types:- The common types of Extreme pressure Additives are those containing sulfur and/or chlorine.

For example, sulfur containing EP additives are:

Dibenzyldisulfide

Sulfurised fatty ester

Sulfurised turpenes and olefins

For example, chlorine containing EP additives are:

Chlorinated paraffins (Trade name Crochlor).

Chlorinated aromatics (Trade name Arochlor)

These EP additives are mainly used in industrial lubrication e.g., metal cutting and metal forming operations, heavy gear lubrication etc.

A very important application of EP lubrication is rear axle of automotive vehicles.

CHAPTER - 2 LITERATURE SURVEY

2.1 SULFUR AND ORGANO-SULFUR ADDITIVES (14)

Sulfur and organosulfur compounds were probably the first well known antiwear and extreme pressure additives. Flowers of sulfur have been added to lubricating oils to increase their load carrying properties for many years. Even today, sulfurized oils are used in certain industrial applications. The most commonly used sulfur additives today are sulfurized fats, sulfurized olefines, and sulfurized turpenes. We shall discuss the following:-

(A) Extreme pressure properties of organo-sulfur compounds.

(B) Antiwear properties of organosulfur compounds.

(A) <u>EP properties</u>:- Organosulfur additives are accepted as the most important additive class for EP lubrication. The EP properties of the following organosulfur compounds will be discussed:-

Dibenzyldisulfide

Diphenyldisulfide

Dinalkyldisulfide

Dibenzylmonosulfide

Ditertbutyl disulfide

Studies[14] on above compounds indicated that the chemical structure of the sulfur compound had a marked effect upon its performance. The disulfides were better than monosulfides and the order of increasing EP activity was diphenyl $<$ di-nbutyl $<$ ditertbutyl $<$ dibenzyl $<$ diallyl. EP performance of a range of dialkyl disulfides decreased with increase in alkyl chain length, whilst increasing alkyl chain length had no effect on EP performance of ditert-alkyldisulfides.

(B) <u>Antiwear Properties</u>:- The antiwear properties of the same organosulfur compounds as in the case of EP properties, were determined by Forbes[14] using four ball antiwear tests.

Organosulfur compounds were found to be poor antiwear additives. The best antiwear properties of organosulfur additives were equivalent to medium antiwear activity of a standard phosphorus compound.

The antiwear properties of disulfides increased along the series di-nbutyl $<$ dialkyl $<$ dibenzyl $<$ diphenyl and monosulfides. Increasing chain length of di-nalkyl disulfides increased their antiwear effectiveness whilst disulfides with branched alkyl groups had inferior properties to their corresponding n-alkyl derivatives.

2.2 MECHANISM OF ANTIWEAR ACTION OF SULFUR COMPOUNDS - METHODS AND RESULTS

The mechanism of action of antiwear and EP additives is not well understood. It is however known that the active element present in the additives forms inorganic films which are responsible for their action. The films themselves are complex mixtures and their formation and nature depends on the sliding conditions. The following paragraphs summarise the recent research findings in this area.

The studies on the mechanism of action of the anti-wear additives can be grouped into two main categories as follows:

1. The static tests and Physicochemical Methods.
2. The Dynamic Tests.

<u>Static-Tests</u>:- In this class the chemical reactivity of EP and antiwear additives, such as sulfur, organosulfur compounds, is studied on metals under static conditions.

Sakurai and Sato[15] studied chemical reactivity of sulfur and chlorine type additives using hot-wire method and related the antiwear action with corrosivity of these additives. They found that corrosion rate was

parabolic and attributed the antiwear action to the formation of barrier films.

Loeser et al[16] used immersion method and found that EP action was due to EP films. ulfur content of their static films increased with immersion time and temperature.

Buckley[17] investigated nature of chemical reaction of oxygen and sulfur with clean iron surface and found iron sulfide films on surface of iron. He found that hydrocarbons such as methyl mercaptan adsorb to iron surface dissociatively with the result that only sulfur remains on the iron surface and hydrocarbons leave the surface. This sulfur adsorbed on iron surface was contributing to the antiwear action according to him.

Dynamic Tests:- Action of additives cannot be explained on the basis of their chemical structure and reactivity alone; interactions with the rubbing surfaces also must be considered. This idea made possible the studies of anti-wear additives in the friction and wear machines.

Rounds[18] studied effect of additives on the friction of steel on steel. He found that additives form surface films of appreciable thickness. He also found that additive concentration controls the film-thickness and nature of the surface topography of the friction surfaces.

Spikes and Cameron[19] found in their friction experiments with dibenzyldisulfide that only a thin EP film was needed to give boundary lubrication.

Nakayama and Sakurai[20] studied chemical wear of copper with n-hexadecane containing elementary sulfur as additive and found that there is an optimum sulfur concentration at which wear rate and friction coefficient are minimum. They found that at lower concentrations adhesive wear was the wear mechanism and at higher concentrations, the wear was by flaking of sulfide films of critical film thickness.

2.3 Electrical Contact Resistance Methods in the Study of Antiwear Additives and Present Approach

Furey[21] developed for the first time a new device to study metallic contact and friction between lubricated sliding surfaces. His system consisted of basically a fixed metal ball loaded against a rotating cylinder. The extent of metallic contact was determined by measuring both the instantaneous and the average resistance between the two surfaces.

Principle:- The electrical resistance was found to oscillate rapidly between an extremely low value and infinity suggesting that metal contact is discontinuous. The average resistance of an oil film is therefore a "time-average", that is, a measure of the percent of the time that metallic contact occurs.

Application:- Using a metallic contact measuring device on above principle, the entire regime from hydrodynamic (no metallic contact) to pure boundary lubrication (continuous metallic contact) can be readily investigated. Furey found that load, speed, mineral oil viscosity, the presence of additives, and operating time were found to be important variables influencing metallic contact. The apparatus is particularly useful in studying the action of

Antiwear and Extreme pressure Additives. The apparatus allows one to measure not only the effectiveness of Antiwear Additives in reducing metallic contact, but also the rate at which they act and the durability of the protective films which may form.

<u>Behaviour of ECR</u>:- Chu and Cameron[21] designed an apparatus to study flow of electric current through lubricated contacts. They made the study of the passage of current, both DC and AC, through elastohydrodynamic lubricated contacts using a 4-ball machine at 175 rpm and with 1" steel balls. At small applied voltages 15 mV, when there is a coherent oil film (i.e. when it is not short circuited by metallic contact) it behaves as an ohmic resistance. The resistance of their oil film varied from 10^4 ohms to 1 ohm. This result leads to percentage metallic contact, and statistical contact method of assessing oil film thickness. At high currents ($\sim$ 1 amp) the current flows by voltage discharge mechanism and has been discussed by the authors.

<u>Recent Findings Using ECR Method</u>:- Recently, Kawamura[23] et.al. studied metallic contact between lubricat surfaces. They applied 0.1 volt between mating surfaces of 4-ball testing machine and observed variation of the voltage owing to the occurence of metallic contact under dynamic conditions. They applied this method to evaluate the effect

of viscosity and antiwear additives on metallic contact. They found that less metallic contact was observed with more viscous oil, that metallic contact was decreased with increase in concentration of antiwear additive (TCP) in the oil, and that metallic contact was different with different additives.

Czichos[34] et al. applied ECR measuring technique to various studies of the contact in partial elastohydrodynamic lubrication and in boundary lubrication. A contact resistance $R_c < 0.1$ ohm was taken to indicate metallic contact, whilst the existance of nonmetallic layer lead to values of $R_c > 1$ k-Ohm. Their R_c values fluctuated in general over some orders of magnitude with pulse durations between microseconds and seconds.

Czichos[34] et al. counted the rate at which R_c is above or below a certain level and then measured the percentage of time during which R_c is above or below this level. From the results, they obtained information on the lubrication mode and on the action of lubricants.

<u>Present Research Work</u>:- The aim of present research "Studies on Wear And Metallic Contact of En 31 Steel Using Elemental Sulfur in White Oil" is directly related to the mechanism of antiwear action of sulfur additives.

There are two possibilities by which an organo-sulfur additive can give antiwear effect in friction and wear applications. Firstly, the additive may help in promoting the hydrodynamic lubrication conditions in a given friction situation and hence reducing the metallic contact and resulting in wear reduction. Secondly, the additive may be chemically reacting with the metal surfaces and forming strongly adherent films and reducing friction and wear due to the presence of such films. It is also possible that the additive may have dual role and hence both these mechanisms may be operating simultaneously. The purpose of the present research is to understand the mechanism whichever is actually operating.

To accomplish the purpose as described above, the experiments and apparatus have been specially designed to make the required studies. Specific features of the studies are as follows:-

Dynamic conditions have been chosen for the experiments with additives, whereas most of the work on additives above so far has been under static conditions and therefore cannot be fully relied upon. The dynamic conditions are achieved by making wear tests in a ball-on-disc wear test rig used in the present research work.

The additive is chosen as elemental sulfur and organosulfur compounds have been avoided. This is because the organosulfur compounds have to dissociate first on the friction surfaces and then the sulfur is released which acts as antiwear additive thus making the antiwear additive system a complex one and direct use of elemental sulfur avoids this complexity. It is well known from static studies that additive concentration plays important role in antiwear action and film formation. Therefore the sulfur is added in different concentrations in present work to find out antiwear domain concentration of sulfur in the lubricant.

The lubricant is chosen as white oil. White oil is a highly refined mineral oil which has mainly straight chain and branched hydrocarbons structure with various side chains. It is not possible to define the molecular structure of the white oil. It can also contain naphthenic hydrocarbons, the aromatics will be negligible. Complex lubricants have been avoided because other additives present in industrial lubricants can interfere with action of main additives of this work and can lead to misguiding results. This is the reason that white oil is used as base oil.

The most important feature of this work is incorporation of Metallic Contact Percentage Indicator which

is an ECR method to study simultaneously the variation of metallic contact as the wear experiment, under particular conditions, progresses. The metallic contact circuit used in present work is basically similar to that used by Czichos et al.[24] and is employing the principle of Furey s Circuit[21].

The approach of measuring metallic contact has been to record "Percentage Metal Contact Vs Time" curve on a pen recorder. This approach is similar to the work of Kawamura et al.[23].

Miscroscopic visual examinations of the surfaces at each stage of wear have been made which, when related with metallic contact Vs wear volume relations, reveal the mechanism of antiwear action of elemental sulfur in white oil, under boundary lubrication conditions.

CHAPTER - 3 APPARATUS [Fig. 3A]

3.1 BALL-ON-DISC WEAR TEST RIG (Fig 3B)

This machine utilised the concept of sliding wear between a fixed steel ball resting on the cylinderical surface of a moving steel disc, the ball being fixed on a beam and lever loaded by this beam. Hence, we obtain a point contact with concentrated load.

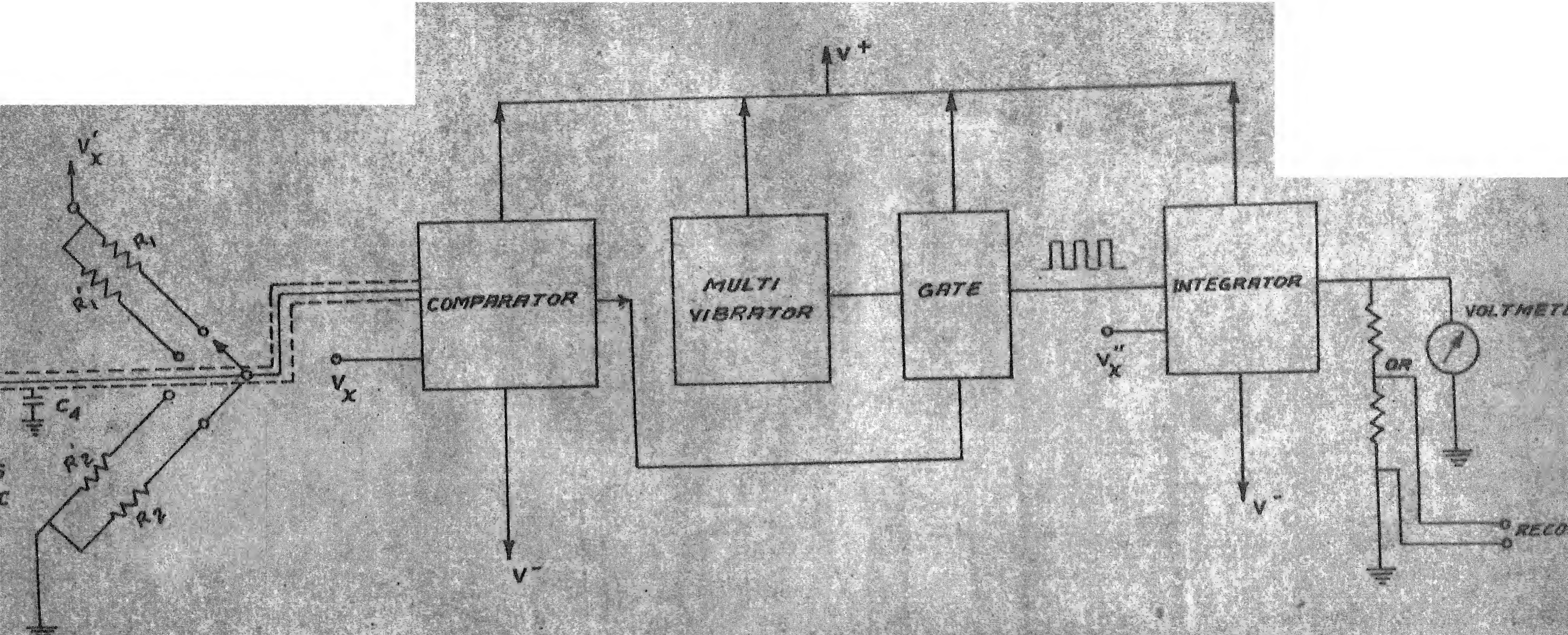
Steel ball is fixed in a ball holder through a ball-chuck. The ball holder is carried by the beam. The beam carries the ball in the centre and a sliding dead weight at one of its ends while the other end of the beam carries a load such that the normal load on the ball, when loaded, is 4 Kgs. Sliding weight is used to balance the beam. The steel ball rests upon the cylinderical surface of a steel disc. The steel disc has a 16 mm bore at the centre and is mounted on the projected end of a hardened steel shaft which has a run-out of not more than 2 microns. The rotating shaft is mounted in a quill and is run at suitable speed with the help of a Prime Mover (Electric Motor) and a speed variator (variable speed drive).

The beam carrying the standard load and the steel ball can be fixed in a desired position and can be slid across the width of the steel disc with the help of a Beam-Adjustable-Lever-Carriage. The beam can be locked in the desired position with the help of a locking nut. By this method, a suitable wear track can be chosen across the cylinderical surface of the steel disc and also the distance between the wear tracks can be adjusted. The disc is fixed in position to prevent any slip between itself and the shaft with the help of a locking nut.

3.2 METALLIC CONTACT PERCENTAGE INDICATOR

This instrument basically estimates percentage of metallic contact duration for a lubricated metallic contact in relative motion. It has been extensively used in this research project. Whenever stable oil film breaks between the rotating disc and the fixed ball under a particular load, a contact occurs between the two. These contacts are of random nature and over a period, this effect is felt in the form of wear scar on the ball and a wear track on the disk.

Figure 3-C shows a block diagram of the metallic contact percentage indicator. The instrument uses two arm resistance bridge; sensitive fast comparator; square wave generator (multivibrator); gate and an integrator. The two-arm resistance bridge operates on a stable d.c. potential, the oil film resistance being one of the arms. At any instant, the comparator swings to either +ve or -ve voltage side depending on whether there is a contact or no contact between the ball and the disc. The square pulses are allowed to go through the gates only if there is a metallic contact. The integrator accepts these square pulses and integrates them averaged over the desired period. The output of the integrator, for which recording facilities are also given, indicates at any time the percentage metal



BLOCK DIAGRAM OF METALLIC CONTACTS PERCENTAGE INDICATOR

FIG - 3 - C

contact duration. Hundred percent metallic contact is defined as a condition where lubricant films do not separate the surfaces. Zero percent means when the ball and the disc are completely separated by an oil film and no metallic contact occurs. Instrument specifications are as following.

Input Sensor:- No special sensor is involved. Oil film between the two metallic parts forms part of the bridge.

Accuracy:- This means the accuracy of the observed metallic contact percentage value taking into account the errors and limits of the comparator and Integrator. The maximum error for full scale value in the present circuit is $\pm 2.5\%$.

Response Time:- This is defined as the output damped for a particular value of time i.e. the time over which the random contacts are averaged. This value is 3 seconds for the present circuit and is adjustable as desired.

Arm Resistance and Metallic Contact:- The two arm resistances are 10 ohms and 1 K-ohm. The 10 ohm resistance is the one across which the oil film resistance will be measured and 1 K-Ohm is standard resistance. A resistance value 1 ohm and below indicates full metallic contact and the resistance greater than 1 ohm indicates no metallic contact.

3.3 LUBRICANT SUPPLY AND TEMPERATURE CONTROL (Fig 3B)

A 150 ml capacity tray made of aluminium has been used as a lubricant bath. In this tray 100 ml of lubricant of desired composition is filled and the tray is placed on the Ball-and-Disc-Wear Test Rig-Bench in such a way that the disc is partially dipping in the lubricant. Due to the rotation of the disc, the lubricant is carried upto the contact zone

In the lubricant tray is immersed an immersion heater of suitable heating capacity (75 watts) for maintaining a temperature of 40°C in the lubricant bath. A contact thermometer is also dipped in the lubricant bath and is connected to the immersion heater through its solid state circuit to control the lubricant temperature. The contact thermometer regulates the power supply to the heater. A separate thermometer capable of reading upto 0.1 Degree is also dipped in the bath along with contact thermometer in order to indicate the actual temperature of the bath. The surrounding temperature was controlled by an air-conditioner to minimise the variations in heat loss.

With all above arrangements, temperature variation was brought within $\pm$ 1/4°C.

3.4 PHOTOGRAPHIC EQUIPMENT

Neophot metallurgical microscope Figure 3-D has been used for visual examination of microstructure of the En 31 steel used in the present work and also to observe the surfaces of the wear scars. Microscopic studies have been particularly useful in understanding the film formation phenomena and etch-resistant areas due to the action of elemental sulfur in white oil. A camera attachment is available for recording visual observations. This microscope has been extensively used to measure the dimensions of the wear scars after different wear-tests, under a standard magnification of 60X, and the observations have been shown in the appendix. The microscopic divisons can be converted into corresponding dimensions in mm by certain multiplication factor, and hence the wear scar measurements are [illegible] made.

A VIEW OF NEOPHOT METALLURGICAL MICROSCOPE

FIGURE : 3 D

A VIEW OF TALYSURF FOR SUBFACE ROUGHNESS DETERMINATION OF DISCS.

FIGURE: 3 E

minimum distance between the datum line and the centre-line which divides the roughness profile area into two halves, the upper half and the lower half about the centre-line itself.

Rough spots on certain discs were detected and some discs had shown tapering when their Talysurf profil was taken. Such discs were rejected.

Hardness Tester:- The hardness of the hardened discs was measured on a Rockwell Hardness Tester Figure 3 - F using Rockwell C scale. The discs with hardness outside the range of 60-65 R_C were rejected. Hence, the metallurgy of the discs was controlled.

4.1 PREPARATION AND PROCEDURE FOR WEAR TESTS

A fresh steel disc of diameter 40$\pm$0.5 mm, hardness R_C 60-65, roughness 1·5–2μ(CLA), is thoroughly cleaned with acetone in Sauxhlet Apparatus and then mounted on the projected end of the shaft on the Ball-And-Disc Wear Test Rig. Disc is locked with the locking nut. The run-out of the disc is checked with a dial gauge and ensured that it is not more than 0 - 2 microns.

The oil bath is placed below the disc properly so that disc is partially immersed in the lubricant and free movement of the disc is possible. Then, the contact thermometer and indicating thermometer both are lowered to dip in the lubricant bath. The oil bath which already contains an immersion heater is heated through the contact thermometer till the temperature is stabilised at 40 $\pm$ 1/4 °C.

A fresh steel ball is mounted in the ball chuck and is fixed in the Wear Test Rig through a ball holder. An insulation tape is used to electrically isolate the steel ball from rest of the apparatus' metallic parts. The ball is connected to Metal Contact Circuit with a lead wire through a nut and screw in the top portion of the ball holder.

Rest of the apparatus is earthed and the common earth point is connected to the shaft through a mercury contact. The mercury contact is made possible by mounting a brass disc on the shaft of the wear test rig and a portion of this brass disc is immersed in a mercury tank placed below the brass disc. This makes possible the earthing of the moving shaft and hence earthing of the steel disc. The other end of the metal contact circuit input is therefore earthed. This means the ball and the disc, when touching through lubricant, form part of the resistance arm across which the voltage and voltage variation and hence the metallic contact is measured.

The zero metal contact and full metal contact are checked by separating the ball and disc, and by touching them respectively and observing the Metal Contact Voltage in the indicator. The output of the metal contact circuit is connected to a multispeed chart recorder.

With all the above arrangements, and setting the recorder at 2 cm per minute speed, apparatus is ready for the wear experiment.

The motor is put on and the disc starts rotating. The rpm is adjusted and checked with speedometer. The beam of the wear test rig is loaded with 4 Kg. load and very slowly the steel ball is placed on the moving disc. After

the desired period of the wear-run, the steel ball is separated from the moving disc by lifting the beam and simultaneously the metal contact recording is stopped. The experiment ends with

- A wear scar on the ball and a wear track on the disc.
- A metal contact curve in the chart recorder.

From wear scar, wear volume can be calculated. The wear scar is measured in the microscope and it looks elliptical in shape. The scar is assumed circular in shape and the average of the major and minor diameters of the elliptical scar gives the diameter of the wear scar circle, for simplifying the calculations.

The fixed parameters of the experiment are the load, the lubricant temperature which is 40^{o}C in all experiments.

The variable parameters are the period of the wear run disc speed, the composition of the lubricant, the nature of the wear run.

4.2 STEP WEAR TESTS AND CONTINUOUS WEAR TESTS

Two types of wear tests have been conducted in this research work, as follows:

1. Step wear Test Runs
2. Continuous Wear Test Runs.

Step Wear Test Runs:- In this category, three wear tests were conducted in steps of half an hour on the same wear-spot and without disturbing the location of the wear scar. First, a half an hour wear test was conducted and the steel ball along with the ball chuck was removed for measuring the wear scar and observing the scar surface topography. Having done so, the same wear scar was again rubbed for half an hour repeating the similar experimental conditions. The increased wear scar was measured and the process repeated for additional third half an hour wear test on the same wear scar. The metal contact was also recorded along with each half an hour step wear test.

These experiments were conducted for pure white oil and for white oil with different sulfur concentrations and respective wear data was obtained.

Continuous Wear Test Runs:- In this category, the tests were conducted in steps of increasing wear period. In each step, corresponding wear scar and metal contact were measured. These experiments were conducted to find wear rate with white oil and varying percentage sulfur in white oil. Also, the relationship between wear volume and metal contact was obtained from continuous wear test runs. In most of the continuous wear test runs, fresh surface of the steel ball was chosen. These are called "Fresh Scar Continuous Wear Tests".

In some experiments, the original wear scar was brought, back into position after microscopic examination and was re-rubbed under similar experimental conditions for all the additional increased wear periods. These are called "Cumulative Continuous Wear Tests".

4.3 WEAR VOLUME AND METAL CONTACT MEASUREMENTS

Wear Volume - The wear scar after a certain wear test is measured under the microscope. The wear scars are elliptical in shape and therefore the average radius of each wear scar is found out. In appendix I, it is discussed how to calculate the wear volume from given wear scar radius when the scar is produced due to the rubbing contact of a fixed steel ball on a rotating steel disc at constant normal load. The plots between wear scar radii Vs wear volumes are also given. From these plots, the wear volume corresponding to a particular scar radius is found out.

Metallic Contact - The strip chart recorder records the percentage metallic contact as a function of time at a standard chart speed of 2 cms/minute. This record for a given wear test is called "Metallic Contact Curve". The area under this cruve gives "metallic contact" for a given wear test and can be calculated. For all the wear tests, their corresponding metallic contact curves areas are found out. Typical metallic contact curves are shown in Fig. 4A.

4.4 MICROSCOPY OF WEAR SCAR SURFACE

After a particular wear test, the surface of the wear scar is examined to study the characteristics of the surface.

The surface can reveal several features. The most important feature is whether the surface is covered with a reation film or not. The scar surface is first seen as it is after the wear test. If it is covered with a barrier film or a reaction film, it will reveal a smooth and scratch free surface under the microscope at suitable magnification. In the second stage of observation, the surface is etched with 4% Nital solution and re-examined. The etching process removes the film and the scratch pattern is then visible which was covered by the film before etching. This technique confirms whether the surface is covered with any barrier film or not.

The corrosive effects of elemental surface in white oil can be also seen under the microscope. The pitted surface and pitted spots indicate corrosive effects.

The scratch pattern and surface roughness can allow distinction between a severely worn surface and a mildly worn surface. Surfaces obtained by dry wear can be distinguished from those obtained by lubricated wear.

4.5 MATERIALS CHARACTERISATION

Lubricant:- Highly refined white oil is chosen as lubricant to form a base oil for lubricated wear tests. White oil is a mineral oil with straight chain and branched chain hydrocarbon structure. The physcochemical properties of the white oil used in present studies were determined and are shown below:

Viscosity in c.s.t.	at 37.8°C :-	64.59
Viscosity in c.s.t.	at 98.9°C :-	7.71
Viscosity Index :-		92
Clevelend Open Cup Flash Point:-		224°C
Ash percent, by wt:-		Practically Nil
Nutralisation Value:- mg KOH/gm.		Practically Nil

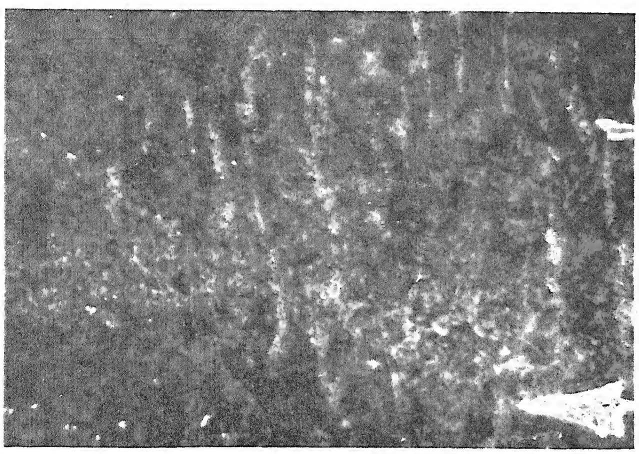
Additive:- I.P. grade powder sulfur was chosen as antiwear additive. The particular sample of sulfur used in present work contained at least 97% sulfur and its various weight fractions were dissolved in the white oil by heating the lubricant and additive mixture until a clear solution was obtained. Heating upto 60°C with constant stirring was found adequate for making a required additive-lubricant solution.

En31 Steel Balls:- The standard fully hardened steel balls with a ball bearing surface finish figure 4 B were used in the wear of En^{31} steel studies. These balls are of 1/2" dia size and are generally used in the fourball testing machine. The Typical microstructure shown in figure 4 B is of the fine tempered martensite with little bainite and carbide rejected to grain boundaries. The typical hardness value of the steel balls was found to be 65 R_c. The roughness can be found by Teleround but it was not available. However the roughness of all the balls was the same of a standard ball bearing type.

En31 Steel Discs:- The steel discs figure 4-C had 40 $\pm$ 0.5 mm as outer diameter and had a 16 mm. inner dia to enable them to fit on the shaft to allow rotation. The thickness of the steel discs was kept 12.7 mm. The typical microstructure shown in figure 4 D is tempered martensite and bainite with some carbide particles. The hardness of the steel discs were confined with the range of 60 R_c - 65 R_c. The surface roughness of the steel discs was measured on the Talysurf 4 and was confined with the range 1.5 - 2.0 (c.l.a) microns.

HARDENED EN 31 STEEL DISCS

FIGURE : 4 C



ICAL MICROSTRUCTURE OF EN 31 STEEL DISCS.

FIGURE : 4 D

The chemical composition of the En[31] steel for both steel balls and steel discs were supplied by the manufacturers as follows:-

Carbon	..	.95 - 1.1%
Silicon	..	.25 - .35%
Chromium	..	1.3 - 1.6%
Manganese	..	.25 - .45%
Phosphorus	..	.025%
Sulfur	..	.025%

The microstructure of both the steel balls and the steel discs were determined in the laboratory in the present studies.

CHAPTER - 5 RESULTS AND DISCUSSION

There were two types of wear tests conducted, namely, Dry Wear Tests and Lubricated Wear Tests.

5.1 Dry Wear Tests Series:- This series includes the results of wear tests in dry contact of rubbing surfaces i.e. no lubricant is externally interposed in the friction surfaces. The steel ball is allowed to wear against the moving steel disc at room temperature and in normal atmospheric conditions.

The series consists of "Cumulative Continuous Wear Tests" and continuous metallic contact measurements are done in all the wear tests.

The purpose of this series was to invesitgate the relationship between Wear Volume Vs Metallic Contact and also to check the relationship of wear volume Vs. time.

5.2 Lubricated Wear tests Series:- This series includes the results of wear tests in which white oil is used as lubricant. Elemental sulfur is added in white oil in various proportions as an antiwear additive. This series consists of two types of wear tests:-

A. Step Wear Tests

B. Continuous Wear Tests.

Step wear tests were conducted to find out the anti-wear domain of elemental sulfur in white oil i.e. the composition range of sulfur in white oil within which sulfur acts as an antiwear additive and results in reduced wear, under the given experimental conditions.

The main purpose of the continuous wear tests at different sulfur levels in white oil was to study the Wear Volume Vs Metallic Contact relationship for white oil and the effect of sulfur addition on this relationship. The continuous wear tests done in this group of experiments are 'The Fresh Scar Continuous Wear Type'.

5.1 DRY WEAR TEST RESULTS

Dry wear experiments have been conducted at two different speeds, remaining conditions being identical in all the experiments. The speeds chosen were 125 rpm and 200 rpm and the results are presented in table (R-1) and (R-2) respectively.

In table (R-1) and (R-2), the wear volumes calculated from the wear scar dimensions of the steel ball have been listed against each sliding distance (time is a measure of distance slid). The metallic contact measurements have been made simultaneously and shown against each wear volume for a given wear test. The last column indicates whether the test is on fresh surfaces chosen for the steel disc and ball or the previous wear scar has been re-rubbed for additional sliding distance on the same wear track of the steel disc.

For the same wear test time, the wear volume at 200 rpm is greater than at 125 rpm by virtue of the fact that greater sliding distance is involved at higher speed and the wear should be obviously greater in this case since the metallic contacts are about equal at both speeds.

The relationship between Wear Volume Vs time (or wear volume Vs. Sliding Distance) has been shown on log-log

TABLE R1

TITLE :- DRY WEAR —"CONTINUOUS WEAR TEST RESULTS"
IDENTICAL EXPERIMENTAL

CONDITIONS

* HUMIDITY 45-65 % NORMAL
* NO LUBRICANT USED
* TEMPERATURE 25°C - 22°C
* HARDENDD En 31 STEEL BALL AGAINEST HARDENED En 31 STEEL DISC
* CUMULATIVE CONTINUOUS WEAR TESTS
* LOAD ON STEEL BALL = 4 kgs.

DISC SPEED ⟶ 125 RPM

EXPERIMENT NO.	DURATION OF WEAR TEST	WEAR SCAR DIA. mm	WEAR VOLUME mm^3	METALLIC CONTACT Cm^2	STARTING CONDITIONS
1-DC m_1	IST. 2 mts.	0.8418	51×10^{-4}	76	FRESH SURFACES
1-DC m_2	IInd. 2 mts	1.1063	1.8×10^{-2}	76+77=153	SAME TRACK SAME SCAR
1-DC m_3	IIIRD. 2 mts.	1.2696	2.6×10^{-2}	153+80 = 233	SAME TRACK SAME SCAR
1-DC m_4	IVth. 2 mts	1.3869	3.8×10^{-2}	233+80=313	SAME TRACK SAME SCAR
2-DC m_4	Ist. 5 mts	1.1017	1.5×10^{-2}	195	FRESH SURFACES
2-DC m_2	IIND. 5 mts.	1.3524	3.4×10^{-2}	195+172=367	SAME TRACK SAME SCAR
3-DC m_1	Ist. 10 mts.	1.4605	4.6×10^{-2}	389	FRESH SURFACES
3-DC m_2	IIND. 10 mts	1.675	8×10^{-2}	389+386=775	SAME TRACK SAME SCAR
4-DC m_1	Ist. 15 mts.	1.2949	2.8×10^{-2}	581	FRESH SURFACES
4-DC m_2	IIND. 15 mts	1.3685	3.6×10^{-2}	581+566=1147	SAME TRACK SAME SCAR
4-DC m_3	IIIRD. 15 mts	1.4145	4×10^{-2}	1147+584=1731	SAME TRACK SAME SCAR
5-DC m_1	IST. 30 mts.	1.3432	3.3×10^{-2}	955	FRESH SURFACES
5-DC m_2	II nd. 30 mts	1.400	3.9×10^{-2}	955+392=1347	SAME TRACK SAME SCAR
5-DC m_3	IIIRD. 30 mts	1.500	5.2×10^{-2}	1347+166=1513	SAME TRACK SAME SCAR

TABLE R2

TITLE :- DRY WEAR – " CONTINUOUS WEAR TEST RESULTS "

IDENTICAL EXPERIMENTAL CONDITIONS :–

* HUMIDITY 45 – 65 % (NORMAL)
* NO LUBRICANT USED
* TEMPERATURE 25 °C – 22 °C
* HARDENED En 31 STEEL BALL AGAINEST HARDENED En 31 STEEL DISC.
* CUMULATIVE CONTINUOUS WEAR TESTS
* LOAD ON STEEL BALL = 4 Kgs.

DISC SPEED ⟶ 200 RPM

EXPERIMENT NO.	DURATION OF WEAR TEST	WEAR SCAR, mm	WEAR VOLUME mm^3	METALLIC CONTACT Cm^2	STARTING CONDITIONS
6 – DC m_1	IST 2 mts	0·989	97.3×10^{-4}	76	FRESH SURFACES
6 – DC m_2	IIND. 2 mts.	1·3639	3.5×10^{-2}	77 + 76 = 153	SAME TRACK SAME SCAR
6 – DC m_3	IIIRD. 2 mts.	1·495	5.1×10^{-2}	154 + 74 = 227	SAME TRACK SAME SCAR
6 – DC m_4	IVth. 2 mts.	1·679	8.1×10^{-2}	227 + 78 = 305	SAME TRACK SAME SCAR
6 – DC m_5	Vth. 2 mts.	1·7687	10×10^{-2}	305 + 76 = 381	SAME TRACK SAME SCAR
7 – DC m_1	ISt. 5 mts.	1·4697	4.8×10^{-2}	195	FRESH SURFACES
7 – DC m_2	IIND. 5 mts.	1·7802	10.2×10^{-2}	195 + 185 = 380	SAME TRACK SAME SCAR
7 – DC m_3	IIIRD. 5 mts.	1·950	15×10^{-2}	380 + 162 = 542	SAME TRAC SAME SCAR
8 – DC m_1	ISt. 10 mts.	1·785	13×10^{-2}	351	FRESH SURFACES
8 – DC m_2	IIND 10 mts.	2·05	18.5×10^{-2}	351 + 388 = 739	SAME TRACK SAME SCAR
8 – DC m_3	IIIRD. 10 mts.	2·15	22.2×10^{-2}	739 + 387 = 1126	SAME TRACK SAME SCAR
9 – DC m_1	IST. 15 mts.	2·150	22.3×10^{-2}	545	FRESH SURFACES
9 – DC m_2	IIND. 15 mts.	2·700	55.1×10^{-2}	545 + 410 = 955	SAME TRACK SAME SCAR
9 – DC m_3	IIIRD. 15 mts.	2·750	59.2×10^{-2}	955 + 581 = 1536	SAME TRACK SAME SCAR
9 – DC m_1	IST. 30 mts.	2·075	19.6×10^{-2}	1176	FRESH SURFACE
10 – DC m_2	IInd 30 mts	2·175	23.4×10^{-2}	1170 + 1112 = 2282	SAME SURFACE SAME SCAR

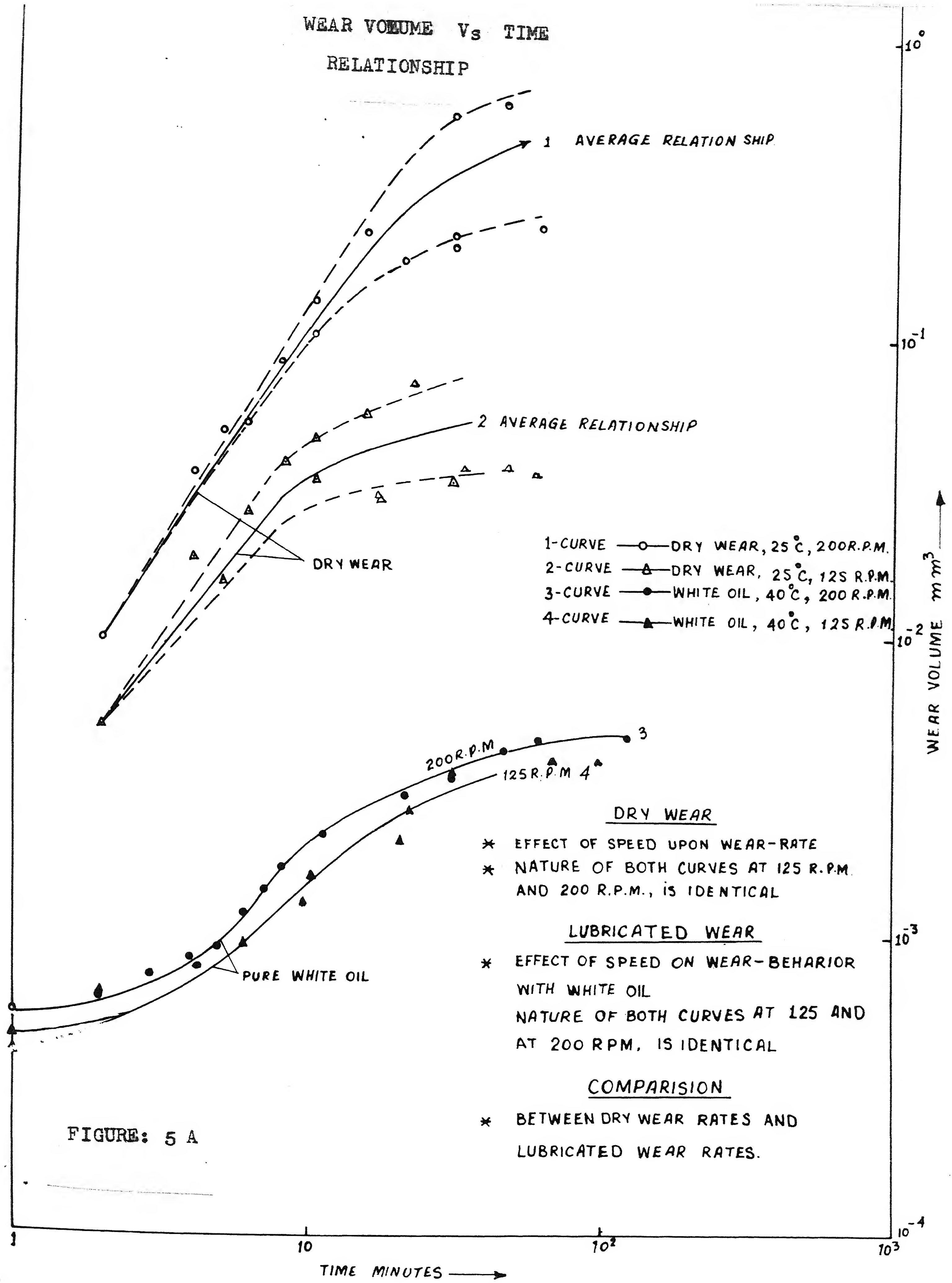

WEAR VOLUME Vs TIME
RELATIONSHIP
1 AVERAGE RELATION SHIP
2 AVERAGE RELATIONSHIP
DRY WEAR
1-CURVE —o— DRY WEAR, 25°C, 200 R.P.M.
2-CURVE —△— DRY WEAR, 25°C, 125 R.P.M.
3-CURVE —●— WHITE OIL, 40°C, 200 R.P.M.
4-CURVE —▲— WHITE OIL, 40°C, 125 R.P.M.
200 R.P.M
125 R.P.M 4
3
PURE WHITE OIL
DRY WEAR
* EFFECT OF SPEED UPON WEAR-RATE
* NATURE OF BOTH CURVES AT 125 R.P.M AND 200 R.P.M., IS IDENTICAL
LUBRICATED WEAR
* EFFECT OF SPEED ON WEAR-BEHARIOR WITH WHITE OIL
NATURE OF BOTH CURVES AT 125 AND AT 200 RPM, IS IDENTICAL
COMPARISION
* BETWEEN DRY WEAR RATES AND LUBRICATED WEAR RATES.
1
10
10^2
10^3
TIME MINUTES
10^0
10^{-1}
10^{-2}
10^{-3}
10^{-4}
WEAR VOLUME mm^3

FIGURE: 5 A

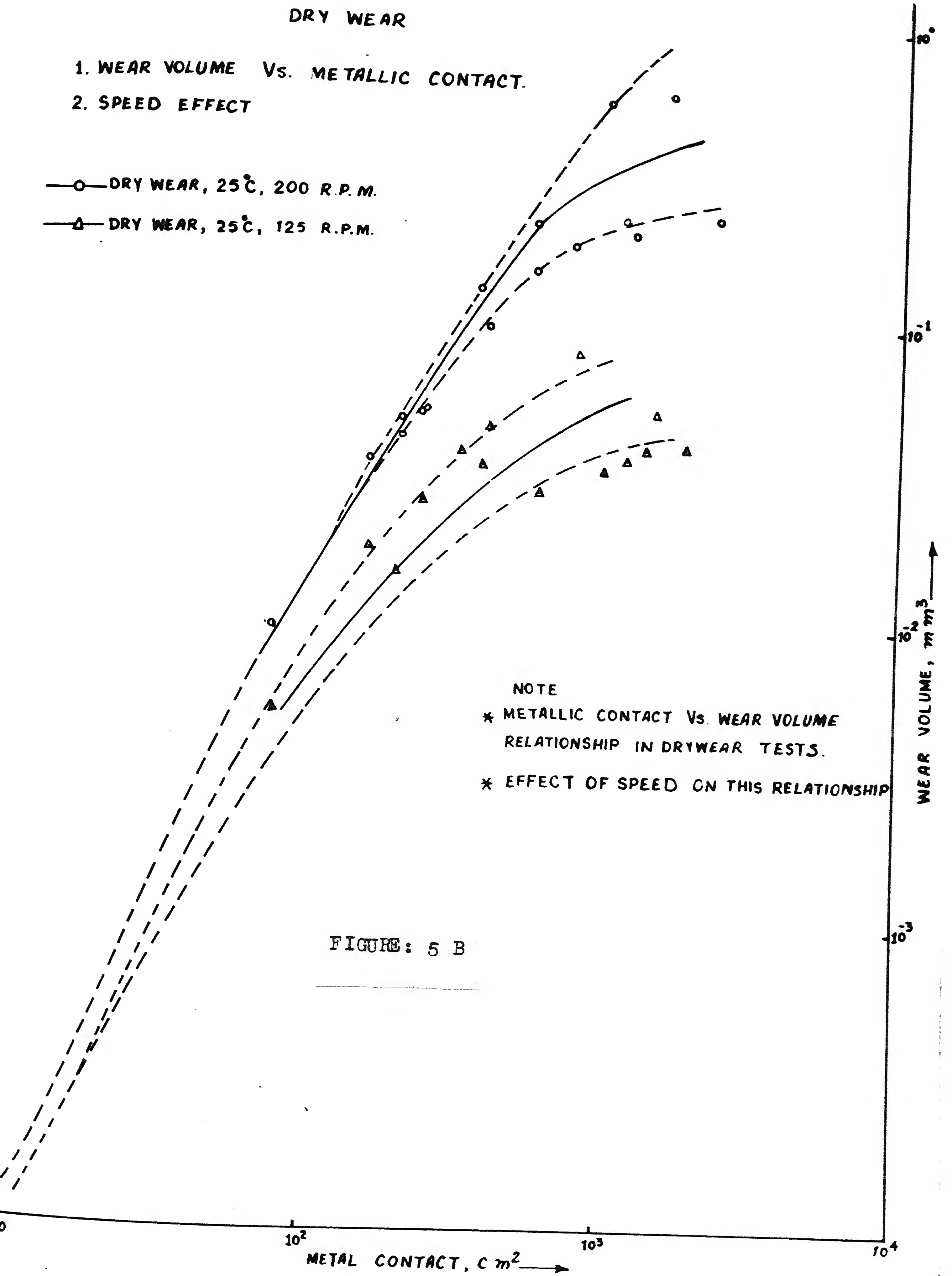

DRY WEAR
1. WEAR VOLUME Vs. METALLIC CONTACT.
2. SPEED EFFECT
DRY WEAR, 25°C, 200 R.P.M.
DRY WEAR, 25°C, 125 R.P.M.
NOTE
* METALLIC CONTACT Vs. WEAR VOLUME RELATIONSHIP IN DRYWEAR TESTS.
* EFFECT OF SPEED ON THIS RELATIONSHIP
WEAR VOLUME, mm³
METAL CONTACT, cm²
10°
10⁻¹
10⁻²
10⁻³
10²
10³
10⁴

FIGURE: 5 B

scale in figure 5-A at both 125 rpm and 200 rpm disc speeds. The wear rate in both the cases is seen to be linear upto a certain time indicating the severe metallic wear stage followed by a decline in the wear rate at both speeds indicating mild wear stage. There is decline in the wear rate at both speeds indicating mild wear stage. The relationship in the severe metallic wear stage is linear whereas the mild wear stage shows a nonlinear wear rate. The non linearity of the mild wear and the decline in wear rate both are attributed [25] to the oxide film formation during the dry wear tests. The oxide films reduce the metal-to-metal contact, reduce the adhesive junction formation process, and thereby decrease the wear-rates.

The relationship between wear volume Vs. metallic contact have been shown on a log-log scale in figure 5-B. There is an initial linear regime in both the cases i.e. 125 and 200 rpm, of dry wear tests. The relationship becomes non-linear after a certain wear has occured at both speeds and the nature of nonlinearity is also similar at both speeds. The non-linearity is explained by the fact that oxidation occurs after certain time in such a way that metal-to-metal contact is reduced which in turn reduces the wear rate and the Wear Volume Vs Metallic Contact relation becomes non-linear. This is attributed to the oxide film formation. The influence of oxides in wear reduction is well known [25] and our results are in agreement with this view.

5.2 LUBRICATED WEAR TEST RESULTS

Lubricated wear experiments are those using pure white oil as lubricant and elemental sulfur in white oil with varying concentrations as additive. The series of experiments conducted under this scheme are as follows:

(A) Lubricated Step Wear Tests

(B) Lubricated Continuous Wear Tests

<u>Lubricated Step Wear Tests</u>:- These tests are done under identifical experimental conditions except for the concentration of elemental sulfur in white oil. The results are shown in table (R-3). The total wear time for each experiment is 90 minutes involving three step wear tests of 30 minutes duration each, at each sulfur concentration in white oil.

It is seen from the data that wear rate and wear volume are both significant functions of elemental sulfur concentration in white oil. The variation is shown in Figure 5-C on linear scale where wear volume for each 30 minutes test is plotted against the corresponding sulfur concentration level.

The figure 5-C indicates that there is an initial increase in the severity of wear with the addition

TABLE: R 3

TITLE :- LUBRICATED WEAR – "STEP WEAR TEST" RESULTS

IDENTICAL EXPERIMENTAL

CONDITIONS :–

* DISC SPEED = 200 R.P.M.
* LOAD ON STEEL BALL = 4 Kgs.
* LUBRICANT TEMPERATURE = $40 \pm 0.25\,^{\circ}C$.
* HARDENED En 31 STEEL BALL AGAINEST HARDENED En 31 STEEL DISC.
* STEP WEAR TESTS CONSIST 3 HALF AN HOUR WEAR TESTS ON THE SAME WEAR-SCAR, WITH SAME TRACK

EXPERIMENT NO.	DURATION OF WEAR TEST	WEAR SCAR DIA. mm	WEAR VOLUME mm^3	LUBRICANT COMPOSITION
1-LS m_1	IST. 30 mts.	0.8004	42.3×10^{-4}	PURE WHITE OIL
1-LS m_2	IIND. 30 mts.	0.8602	55.6×10^{-4}	PURE WHITE OIL
1-LS m_3	IIIRD. 30 mts.	0.9223	73.4×10^{-4}	PURE WHITE OIL
2-LS m_1	IST. 30 mts	1.0419	104×10^{-4}	0.1 % S IN WHITE OIL
2-LS m_2	IIND. 30 mts.	1.0787	107×10^{-4}	0.1 % S IN WHITE OIL
2-LS m_3	IIIRD. 30 mts.	1.1247	150×10^{-4}	0.1% S IN WHITE OIL
3-LS m_1	IST. 30 mts	0.8947	65.2×10^{-4}	0.2% S IN WHITE OIL
3-LS m_2	IIND. 30 mts	0.9039	67.5×10^{-4}	0.2% S IN WHITE OIL
3-LS m_3	IIIRD. 30 mts.	0.9131	70.6×10^{-4}	0.2% S IN WHITE OIL
4-LS m_1	IST. 30 mts.	0.7475	31.4×10^{-4}	0.3% S IN WHITE OIL
4-LS m_2	IIND. 30 mts.	0.7521	32.3×10^{-4}	0.3% S IN WHITE OIL
4-LS m_3	IIIRD. 30 mts.	0.7521	32.3×10^{-4}	0.3% S WHITE OIL
5-LS m_1	IST. 30 mts.	0.6854	22.6×10^{-4}	0.5% S IN WHITE OIL
5-LS m_2	IIND. 30 mts.	0.6900	23.2×10^{-4}	0.5% S IN WHITE OIL
5-LS m_3	IIIRD. 30 mts.	0.6946	23.7×10^{-4}	0.5% S IN WHITE OIL
6-LS m_1	IST. 30 mts.	0.7245	28×10^{-4}	0.8% S IN WHITE OIL
6-LS m_2	IIND. 30 mts.	0.7245	28×10^{-4}	0.8% S IN WHITE OIL
6-LS m_3	IIIRD. 30 mts.	0.7314	28.8×10^{-4}	0.8% S IN WHITE OIL
7-LS m_1	IST. 30 mts.	0.7774	38×10^{-4}	1% S IN WHITE OIL
7-LS m_2	IIND 30 mts	0.8027	42.3×10^{-4}	1% S IN WHITE OIL
7-LS m_3	IIIRD 30 mts.	0.8165	45.5×10^{-4}	1% S IN WHITE OIL
8-LS m_1	IST. 30 mts.	0.9568	85×10^{-4}	2% S IN WHITE OIL
8-LS m_2	IIND. 30 mts.	0.9959	100×10^{-4}	2% S IN WHITE OIL
8-LS m_3	IIIRD. 30 mts.	1.0097	105×10^{-4}	2% S IN WHITE OIL

EFFECT OF SULFUR CONCENTRATION ON WEAR VOLUME

LUBRICATED - STEP - WEAR - TESTS

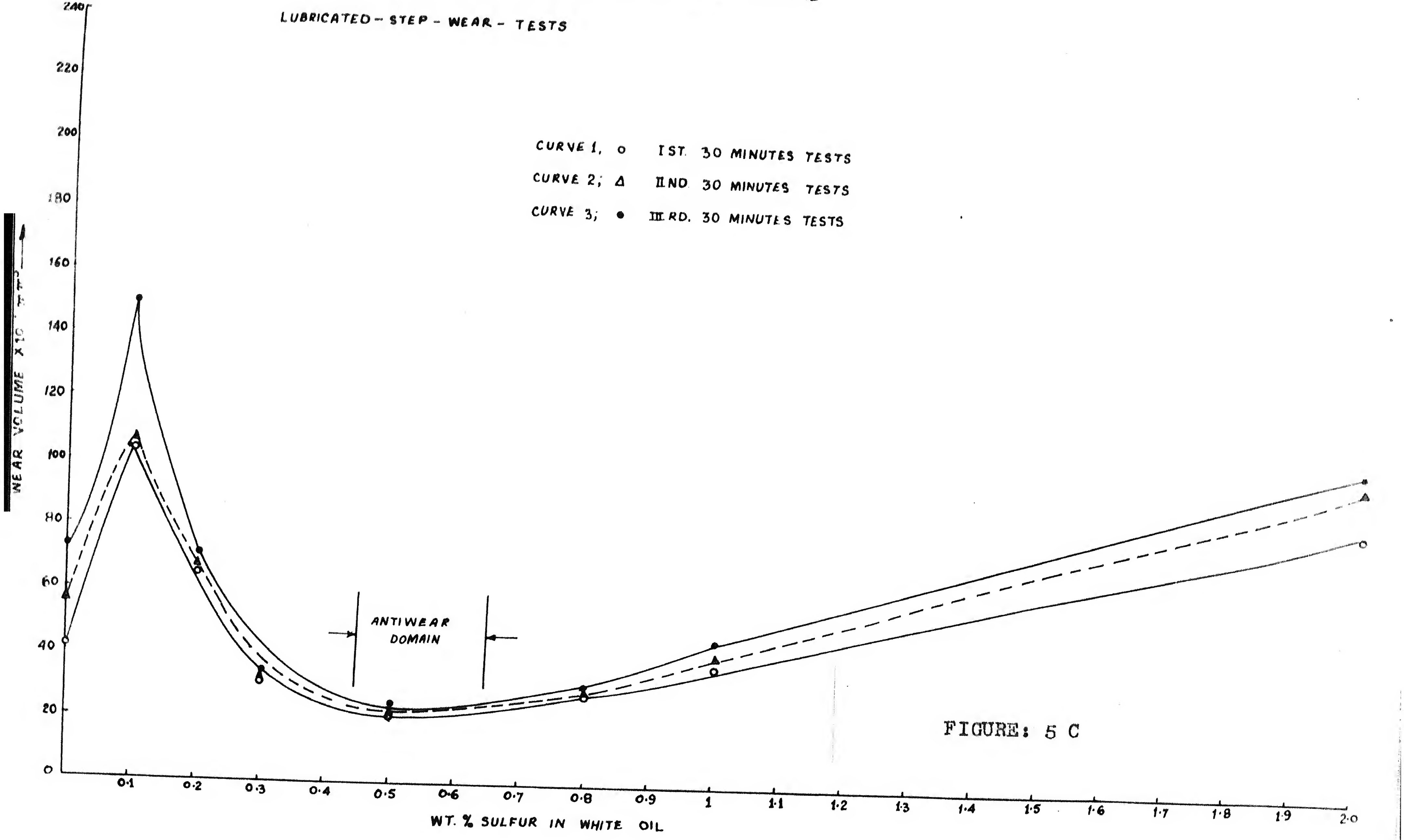


FIGURE: 5 C

of 0.1% sulfur. As the sulfur level increases further, there is decrease in the wear volume until sulfur reaches 0.5 wt.% concentration at which there is seen to be minimum wear. Again for sulfur levels greater than 0.5 wt.%, there is rise in the wear rate with further increase in sulfur concentration which continues to 2 wt.% sulfur. The point to be noted here is that the increase in wear volume with increase in sulfur above 0.5 wt.% is rather slow and the rise in wear volume with decrease in sulfur below 0.5 wt.% S is quite rapid upto 0.1 wt.% S.

The wear volume with 0.1% S and 0.2% S is higher than white oil. With low concentrations of 0.1%, probably there is no effective film formation by sulfur. At the same time, this level of sulfur may reduce oxidation of surfaces as sulfur competes with oxygen for the surface. With reduced surface oxidation the adhesive wear should be be higher than in pure white oil. With 0.2% S, the film effects increase and the wear becomes lower than 0.1% but still higher than that of white oil alone.

At 0.3% S, the film effects are such that wear is lower than white oil. With increase in sulfur concentration when it reaches 0.5 wt.% S, there is minimum wear shown by the data in figure 5-C. This means that film effects are highest at 0.5 wt.% S level resulting in lowest wear volume.

With further increase in sulfur above 0.5 wt.%, there is gradual increase in wear volume as shown by all the three curves. This is probably due to the "film-flaking" effect of the reaction film. This behaviour continues upto 2 Wt.% sulfur concentration in white oil.

The rapid decrease, from 0.1 Wt.% S to 0.5 Wt.% S, of the wear volume indicates that film formation phenomena is much more sensitive with regard to change in additive (sulfur) concentration in the lubricant(white oil) as compared to film flaking phenomena observed with higher sulfur concentration in the film wear stage from 0.65 to 2 wt.% sulfur. This means to say that film flaking process is less-composition-sensitive with regard to change in additive concentration.

The film flaking phenomena has also been observed by Nakayama and Sakurai[20] in n-hexadecane containing elemental sulfur as additive in their studies on chemical wear of copper.

Sulfur is found to be poor antiwear additive. This is clear by comparing the wear volume with white oil and wear volume with 0.5 wt.% sulfur solution in white oil where the factor of wear volume reduction is only two. However, it is to be noted that sulfur does act as an antiwear additive. This finding is in agreement with studies on

sulfur and organosulfur compounds by Forbes[14] who also found that sulfur and organosulfur compounds are mild antiwear additive.

From figure 5-C, the antiwear domain of sulfur is found to be approximately 0.45 to 0.65 wt.% S, and within this range, sulfur acts as an antiwear additive in white oil under the given set of experimental conditions. To find the antiwear domain of sulfur in white oil was the primary objective of conducting the step wear experiments.

Lubricated Continuous Wear Tests:- The continuous wear tests in differently lubricated conditions were designed on the basis of step wear test results. Step wear test results have already indicated that:-

(1) High wear domain exists at 0.1 wt.% S and also at 1 Wt.% S concentrations in white oil.

(2) Antiwear domain exists in the range 0.45 - 0.65 wt.% sulfur in white oil.

Hence, three different additive concentrations of sulfur in white oil were selected for further studies to understand the mechanism of antiwear action of elemental sulfur. These were to be 0.1 Wt.%, 0.5 Wt.%, and 1 Wt.% sulfur solutions in white oil.

Continuous wear with pure white oil was also studied for the purpose of reference oil to compare the behaviour of additive (sulfur) in high wear domains and in the anti-wear domain.

Continuous wear tests were all conducted at 200 rpm and under other identical experimental conditions as shown in the tabulated results.

Continuous wear tests of pure white oil were also conducted at lower disc-speed of 125 rpm to see the effect of speed on various relationships.

Metallic contact measurements were made throughout in all the experiments of continuous wear type.

The tabulated results are shown as follows:

Table (R-4) shows wear test results for white oil at 125 rpm disc speed.

Table (R-5) shows wear test results for white oil at 200 rpm disc speed.

Table (R 6) shows wear test results for white oil containing 0.1 wt.% sulfur as additive, at 200 rpm disc speed.

Table (R-7) shows wear test results for white oil containing 0.5 wt.% sulfur as additive, at 200 rpm disc speed.

Table (R-8) shows wear test results for white oil containing 1 wt.% sulfur as additive, at 200 rpm disc speed.

TABLE: R4

LUBRICATED WEAR–"CONTINUOUS WEAR TEST RESULTS"

IDENTICAL EXPERIMENTAL CONDITIONS :–

* LUBRICANT IS PURE WHITE OIL
* LUBRICANT TEMPERATURE $40 \pm 0.25^\circ C$.
* HARDENED En 31 STEEL BALL AGAINEST HARDENED En 31 STEEL DISC.
* CUMULATIVE CONTINUOUS WEAR TESTS
* LOAD ON STEEL BALL = 4 Kgs.

DISC SPEED ⟶ 125 RPM.

EXPERIMENT NO.	DURATION OF WEAR TEST	WEAR SCAR DIA. mm	WEAR VOLUME mm^3	METALLIC CONTACT Cm^2	STARTING CONDITIONS
1-LC m_1	Ist 30 Secs	0.41055	2.9×10^{-4}	6	FRESH SURFACES
1-LC m_2	IInd 30 Secs	0.48875	5.8×10^{-4}	6+5 = 11	SAME TRACK SAME SCAR
1-LC m_3	IIIRD. 30 Secs.	0.53935	8.6×10^{-4}	11+6 = 17	SAME TRACK SAME SCAR
1-LC m_4	IVth. 30 secs.	0.5428	8.8×10^{-4}	17+5 = 22	SAME TRACK SAME SCAR
2-LC m_1	Ist. 2 mts.	0.50945	6.4×10^{-4}	27	FRESH SURFACES
2-LC m_2	IInd. 4 mts.	0.5612	9.9×10^{-4}	27+47 = 74	SAME TRACK SAME SCAR
2-LC m_3	IIIRD. 4 mts.	0.6003	12.9×10^{-4}	74+35 = 109	SAME TRACK SAME SCAR
3-LC m_1	Ist. 10 mts	0.6325	16.2×10^{-4}	99	FRESH SURFACES
3-LC m_2	IIND. 10 mts	0.71415	26.3×10^{-4}	99+95 = 194	SAME SCAR SAME TRACK
3-LC m_3	IIIRD. 10 mts.	0.7728	37.2×10^{-4}	194+91 = 285	SAME TRACK SAME SCAR
4-LC m_1	IST. 20 mts.	0.67735	21.5×10^{-4}	168	FRESH SURFACES
4-LC m_2	IIND. 10 mts.	0.76245	35.5×10^{-4}	168+63 = 231	SAME TRACK SAME SCAR
5-LC m_1	IST. 30 mts.	0.7728	37.3×10^{-4}	454	FRESH SURFACES
5-LC m_2	IInd. 30 mts	0.8671	40.3×10^{-4}	454+222 = 676	SAME SCAR SAME TRACK
5-LC m_3	IIIRD. 30 mts	0.8855	41.6×10^{-4}	676+35 = 711	SAME SCAR SAME TRACK.

TABLE R5

TITLE :- LUBRICATED WEAR – "CONTINUOUS WEAR TEST" RESULTS IDENTICAL EXPERIMENTAL

CONDITIONS :-

* DISC SPEED = 200 R.P.M.
* LOAD ON STEEL BALL = 4 Kgs
* LUBRICANT TEMPERATURE = 40 ± 0.25 °C
* HARDENED En 31 STEEL BALL AGAINEST HADDENED En 31 STEEL DISC

LUBRICANT :- PURE WHITE OIL

EXPERIMENT NO.	DURATION OF WEAR TEST	WEAR SCAR DIA. mm	WEAR VOLUME mm^3	METAL CONTACT cm^2	STARTING CONDITION
1-LC	1 mts.	0.4876	5.7×10^{-4}	42	FRESH SURFACES
2-LC	2 mts.	0.4991	6.3×10^{-4}	81	FRESH SURFACES
3-LC	3 mts.	0.5221	7.5×10^{-4}	119	FRESH SURFACES
4-LC	4 mts.	0.53705	8.5×10^{-4}	154	FRESH SURFACES
5-LC	4 mts.	0.5244	7.7×10^{-4}	157	FRESH SURFACES
6-LC	5 mts.	0.54855	9.1×10^{-4}	161	FRESH SURFACES
7-LC	6 mts	0.5911	12.1×10^{-4}	223	FRESH SURFACES
8-LC	7 mts	0.61525	14.4×10^{-4}	262	FRESH SURFACES
9-LC	8 mts	0.6371	16.8×10^{-4}	303	FRESH SURFACES
10 LC	11 mts	0.6831	22.2×10^{-4}	395	FRESH SURFACES
11-LC	21 mts.	0.7406	30.4×10^{-4}	623	FRESH SURFACES
12-LC	30 mts	0.7567	33.6×10^{-4}	675	FRESH SURFACES
13-LC	45 mts.	0.8004	42.3×10^{-4}	847	FRESH SURFACES
14-LC	60 mts.	0.8211	46.3×10^{-4}	777	FRESH SURFACES
15-LC	120 mts.	0.81765	45.5×10^{-4}	716	FRESH SURFACES.

TABLE R6

TITLE :- LUBRICATED WEAR – "CONTINUOUS WEAR TEST RESULTS"

IDENTICAL EXPERIMENTAL

CONDITIONS :-
* DISC SPEED = 200 R.P.M.
* LOAD ON STEEL BALL = 4 Kgs.
* LUBRICANT TEMPERATURE = 40 ± 0.25°C
* HARDENED En 31 STEEL BALL AGAINEST HARDENED En 31 STEEL DISC.

LUBRICANT :- WHITE OIL CONTAINING 0.1% SULFUR AS ADDITIVE.

EXPERIMENT	DURATION OF WEAR TEST	WEAR SCAR DIA. mm.	WEAR VOL., mm^3	METALLIC CONTACT cm^2	STARTING CONDITION
16-LC	1 mt.	0.4968	6.2×10^{-4}	38.5	FRESH SURFACES
17-LC	1 mt.	0.5014	6.3×10^{-4}	38.8	FRESH SURFACES
18-LC	2 mts.	0.5819	11.4×10^{-4}	79	FRESH SURFACES
19-LC	3 mts.	0.6693	20.7×10^{-4}	118	FRESH SURFACES
20-LC	4 mts.	0.7130	26.2×10^{-4}	156	FRESH SURFACES
21-LC	5 mts.	0.7866	39.8×10^{-4}	190	FRESH SURFACES
22-LC	6 mts.	0.7981	42×10^{-4}	225	FRESH SURFACES
23-LC	7 mts.	0.8280	47.8×10^{-4}	269	FRESH SURFACES
24-LC	8 mts.	0.8855	62.3×10^{-4}	306	FRESH SURFACES
25-LC	11 mts.	0.9131	70.6×10^{-4}	428	FRESH SURFACES
26-LC	21 mts.	1.0557	1.2×10^{-4}	814	FRESH SURFACES
27-LC	30 mts.	1.1385	1.55×10^{-2}	1116	FRESH SURFACES
28-LC	60 mts.	1.1707	1.9×10^{-2}	2071	FRESH SURFACES
29-LC	120 mts.	1.4651	4.7×10^{-2}	4104	FRESH SURFACES

TABLE R7

TITLE :- LUBRICATED WEAR–"CONTINUOUS WEAR TEST" RESULTS

IDENTICAL EXPERIMENTAL

CONDITIONS :-

* DISC SPEED = 200 R.P.M.
* LOAD ON STEEL BALL = 4 Kgs.
* LUBRICANT TEMPERATURE = 40 ± 0·25 °C.
* HARDENED En 31 STEEL BALL AGAINST HARDENED En 31 STEEL DISC.

LUBRICANT :- WHITE OIL CONTAINING 0·5 % SULFUR AS ADDITIVE.

EXPERIMENT NO.	DURATION OF WEAR TEST	WEAR SCAR DIA., mm	WEAR VOL., mm^3	METALLIC CONTACT cm^2	STARTING CONDITION
30-LC	1 mt.	0·4784	$5{\cdot}2 \times 10^{-4}$	38	FRESH SURFACES
31-LC	2 mts.	0.5566	$9{\cdot}6 \times 10^{-4}$	79	FRESH SURFACES
32-LC	3 mts.	0.5658	$10{\cdot}2 \times 10^{-4}$	115	FRESH SURFACES
33-LC	4 mts	0·5888	$11{\cdot}9 \times 10^{-4}$	156	FRESH SURFACES
34-LC	5 mts.	0.6877	$22{\cdot}8 \times 10^{-4}$	200	FRESH SURFACES
35-LC	6 mts.	0·6992	$24{\cdot}3 \times 10^{-4}$	231	FRESH SURFACES
36-LC	7 mts.	0·6394	17×10^{-4}	260	FRESH SURFACES
37-LC	8 mts.	0·6808	22×10^{-4}	289	FRESH SURFACES
38-LC	11 mts.	0·6693	$20{\cdot}5 \times 10^{-4}$	386	FRESH SURFACES
39-LC	21 mts.	0·7153	$26{\cdot}5 \times 10^{-4}$	684	FRESH SURFACES
40-LC	60 mts.	0·7659	36×10^{-4}	1238	FRESH SURFACES
41-LC	120 mts.	0·8326	$48{\cdot}8 \times 10^{-4}$	1872	FRESH SURFACES

TABLE → R8

TITLE :- LUBRICATED WEAR – "CONTINUOUS WEAR TEST" RESULTS

IDENTICAL EXPERIMENTAL CONDITIONS :-

* DISC SPEED = 200 R.P.M.
* LOAD ON STEEL BALL = 4 Kgs.
* LUBRICANT TEMPERATURE = 40 ± 0·25 °C
* HARDENED En 31 STEEL BALL AGAINST HARDENED En 31 STEEL DISC

LUBRICANT :- WHITE OIL CONTAINING 1 % SULFUR AS ADDITIVE

EXPERIMENT NO.	DURATION OF WEAR TEST	WEAR SCAR, mm	WEAR VOL., mm^3	METALLIC CONTACT cm^2	STARTING CONDITION
42-LC	1 mt.	0·4370	3.6×10^{-4}	39	FRESH SURFACES
43-LC	2 mts.	0·5037	6.3×10^{-4}	77	FRESH SURFACES
44-LC	3 mts.	0·5589	9.8×10^{-4}	113	FRESH SURFACES
45-LC	4 mts.	0·6187	14.8×10^{-4}	152	FRESH SURFACES
46-LC	5 mts.	0·6555	19×10^{-4}	188	FRESH SURFACES
47-LC	6 mts.	0·7268	28.2×10^{-4}	231	FRESH SURFACES
48-LC	7 mts.	0·7360	29.6×10^{-4}	261	FRESH SURFACES
49-LC	8 mts.	0·7475	31.5×10^{-4}	301	FRESH SURFACES
50-LC	11 mts.	0·8970	65.9×10^{-4}	427	FRESH SURFACES
51-LC	21 mts.	1·0695	1.2×10^{-2}	766	FRESH SURFACES
52-LC	32 mts.	1·0074	105×10^{-4}	1229	FRESH SURFACES
53-LC	49 mts.	1·1684	1.8×10^{-4}	1700	FRESH SURFACES
54-LC	60 mts.	1·0051	104×10^{-4}	1488	FRESH SURFACES
55-LC	60 mts.	0·9591	85.7×10^{-4}	1430	FRESH SUFACES
56-LC	120 mts.	0·9798	92.8×10^{-4}	2027	FRESH SURFACES

The conditions and alteration in the same if any have been indicated wherever necessary.

The tables show the duration of each wear test and corresponding wear volume and metallic contact. The starting surface conditions, whether fresh surfaces or otherwise have been indicated in the last column of each table of results.

Results with Pure White Oil:- The following relationship have been studied using pure white oil as lubricant.

* Wear Volume Vs time behaviour shown by figure 5A.
* Wear Volume Vs Metallic Contact behaviour shown by figure 5-D.

The wear volume Vs time relation:- Figure 5-A indicates that the trend is same for both speeds i.e. 125 rpm and 200 rpm disc speeds. There is an initial low wear rate due to running in of the surfaces followed up by a transition to high wear rate after certain sliding distance (or time). Such relationships as in the present case of white oil have been observed with cetane after ten minutes at low stress of 600 psi[13]. The increased wear rates in such cases of short wear time at low stresses and moderate speeds have been attributed to surface fatigue effects. Transition from a constant low wear rate to high wear rate is suggested to be due to the fatigue of the surface by repeated stress cycling. Followed by high wear rate domain of white oil,

there is decline in the wear rate which is attributed to the establishment of hydrodynamic lubrication conditions.

It needs mentioning that the lubricated wear rate behaviour with white oil is, on comparison, found to be different from dry wear tests shown on the same plot in figure 5-A. The differences are as follows.

The dry wear tests show initial constant wear rates of severe metallic wear type followed up by mild wear stage due to oxidation. But the white oil tests show a slow initial wear rate due to running-in and this is followed up with transition to high wear due to surface fatigue by repeated cyclic hertzian stresses.

The dry wear tests show a continuously increasing wear with increase in sliding distance whereas with white oil, wear stops after certain sliding distance beyond the transition to high wear due to fatigue at both 125 and 200 rpm because of the establishment of fluid film between friction surfaces and this is also confirmed by metallic contact curves which indicate continuous high percentage metallic contact in the case of dry wear tests and almost zero metallic contact in the case of white oil tests after

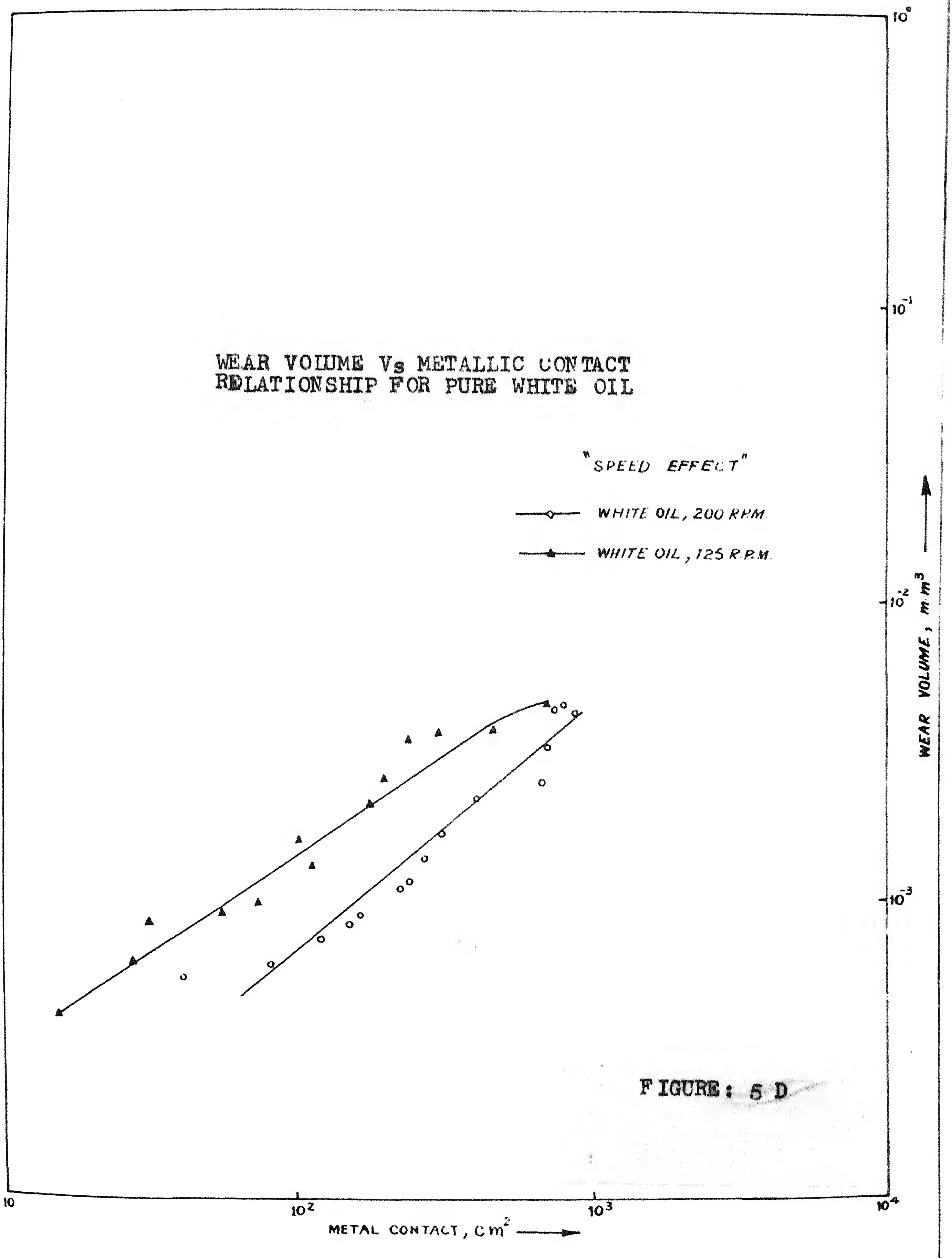

WEAR VOLUME Vs METALLIC CONTACT
RELATIONSHIP FOR PURE WHITE OIL
"SPEED EFFECT"
WHITE OIL, 200 RPM.
WHITE OIL, 125 R.P.M.
WEAR VOLUME, m.m³
10⁰
10⁻¹
10⁻²
10⁻³
10
10²
10³
10⁴
METAL CONTACT, cm²

FIGURE: 5 D

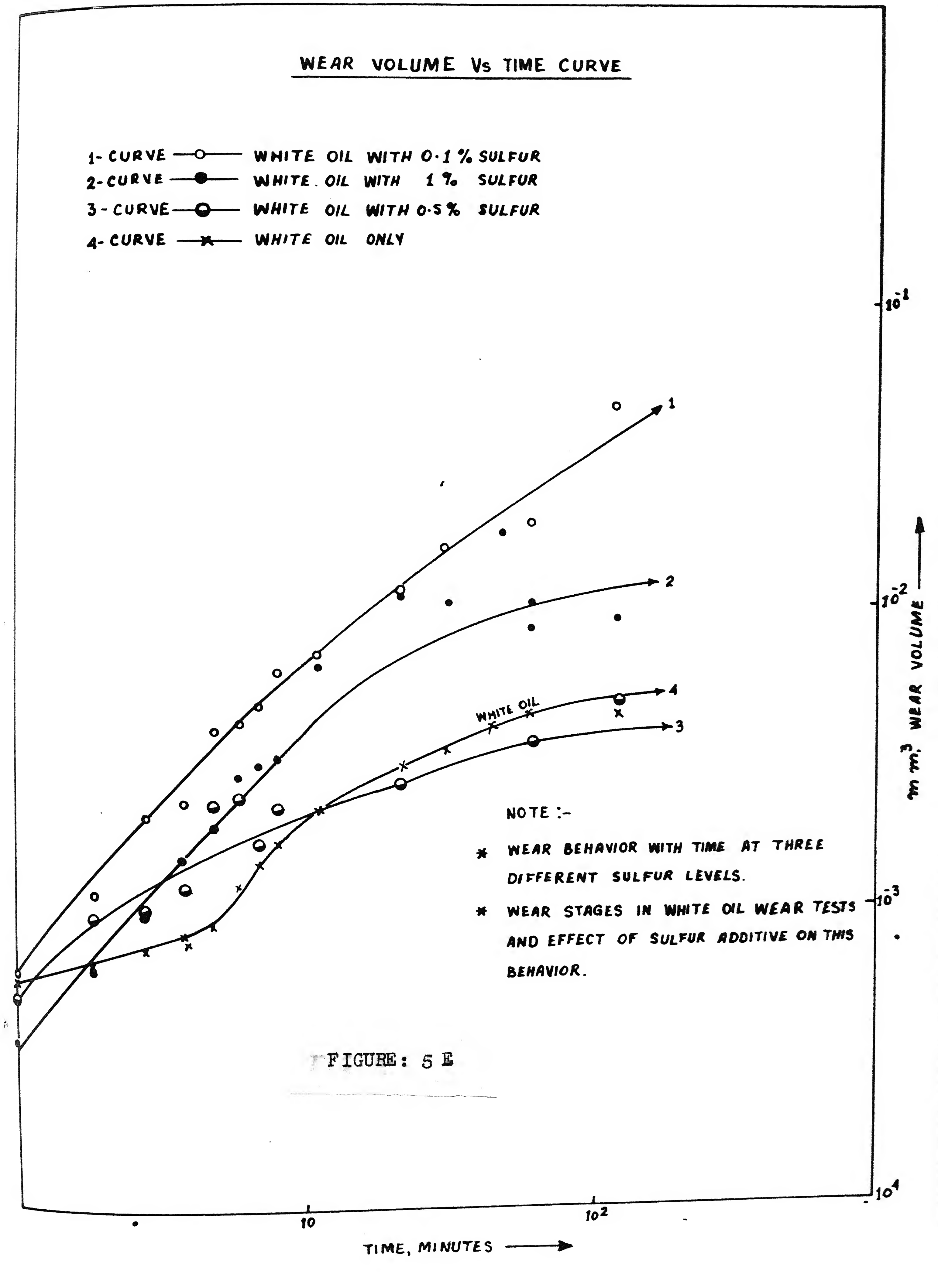

WEAR VOLUME Vs TIME CURVE
1- CURVE WHITE OIL WITH 0·1 % SULFUR
2- CURVE WHITE OIL WITH 1 % SULFUR
3 - CURVE WHITE OIL WITH 0·5 % SULFUR
4- CURVE WHITE OIL ONLY
1
2
WHITE OIL
4
3
NOTE :-
* WEAR BEHAVIOR WITH TIME AT THREE DIFFERENT SULFUR LEVELS.
* WEAR STAGES IN WHITE OIL WEAR TESTS AND EFFECT OF SULFUR ADDITIVE ON THIS BEHAVIOR.
10^{-1}
10^{-2}
10^{-3}
10^{4}
mm^3. WEAR VOLUME
10
10^2
TIME, MINUTES

FIGURE: 5 E

certain sliding distance. The mechanism of fluid film lubrication has already been discussed (Chapter - One, 1.7F).

The Wear Volume Vs Metallic Contact relation:- Figure 5-D indicates the Wear Volume Vs Metallic Contact relationship for white oil at two speeds 125 and 200 rpm.

There seems to be approximately linear relationship between wear volume and metallic contact. It is also seen that for the same metallic contact, wear volume at 125 rpm is greater than at 200 rpm. This is due to the increased hydrod ynamic effects at higher speed.

Results with Additive (Sulfur) in white oil:- The following relationships have been studied using different concentrations of sulfur in white oil.

* Wear Volume Vs Time behaviour shown by figure 5-E.
* Wear Volume Vs Metallic Contact behaviour shown by figure 5-F.

The Wear Volume Vs Time relationship:- Figure 5-E The initial increase in wear rates with sulfur addition has already been discussed in step wear test series.

The nonlinear portion of the curves, figure 5-E, can be explained as follows. The decrease in wear rate can be either due to hydrodynamic effects on lubricant or due to reaction film formation on friction surfaces. To

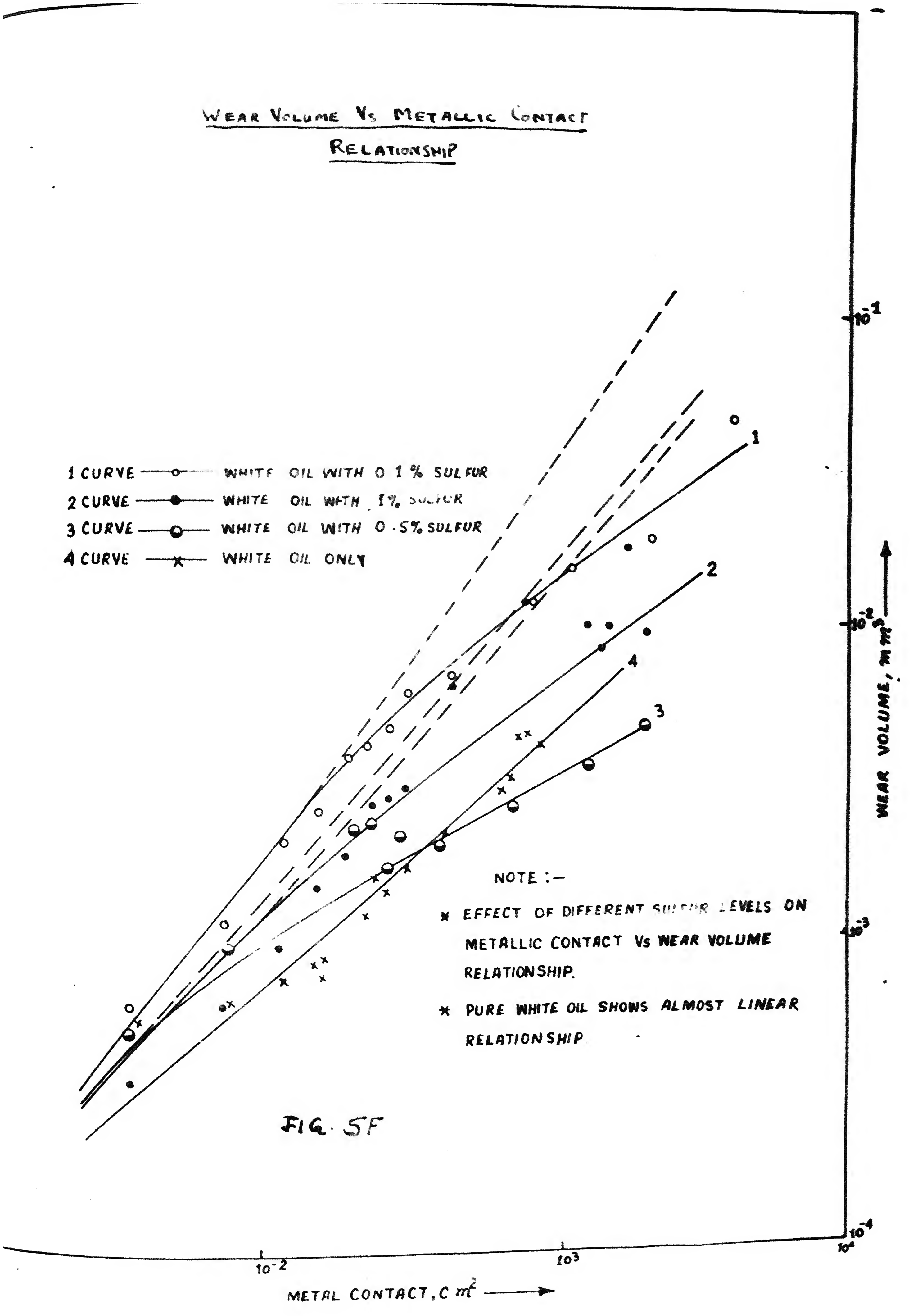
To identify which of the two factors is dominating, it is essential to study wear volume Vs Metallic contact relationship which follows.

<u>The Wear Volume Vs Metallic Contact relationship</u>:-
Figure (5F) shows this relationship for various sulfur concentrations. The relationship for pure white oil as lubricant is also shown for comparison.

Addition of sulfur seems to modify the relationship of wear volume and metallic contact. There is an initial linear relation regime followed up by a nonlinear relation regime.

The initial linear regime of Wear Volume Vs Metallic Contact suggests that wear volume and metallic contact may be quantitatively related.

The linear portions have different slopes. Maximum slope is obtained with 0.1% Sulfur indicating high wear rate upto a wear volume of 1.8×10^{-3} mm^3 and metallic contact of 140 cm^2. Then we observe that 1% sulfur curve has an intermediate slope, the relation in this case is linear upto a wear volume of 1.2×10^{-3} mm^3 and metallic contact of 100 cm^2. The 0.5% S - curve is linear only upto wear volume of 5×10^{-4} mm^3 and metallic contact of 40 cm^2. This behaviour is in good agreement with step wear tests already discussed.


WEAR VOLUME Vs METALLIC CONTACT RELATIONSHIP
1 CURVE WHITE OIL WITH 0 1 % SULFUR
2 CURVE WHITE OIL WITH 1% SULFUR
3 CURVE WHITE OIL WITH 0 .5% SULFUR
4 CURVE WHITE OIL ONLY
NOTE :-
* EFFECT OF DIFFERENT SULFUR LEVELS ON METALLIC CONTACT Vs WEAR VOLUME RELATIONSHIP.
* PURE WHITE OIL SHOWS ALMOST LINEAR RELATIONSHIP
1
2
3
4
WEAR VOLUME, mm³
10⁻¹
10⁻²
10⁻³
10⁻⁴
METAL CONTACT, C m²
10⁻²
10³
10⁴
FIG. 5F

The linear portion of wear volume Vs Metallic Contact relationship is followed up by a nonlinear regime. The levelling off of the curves may not be due to enhanced hydrodynamic effects and points to the possibility of wear resistant reaction films. It is considered that this is not due to hydrodynamic effects because with pure white oil continuous wear tests, there was no levelling off in its curve though significant hydrodynamic effects are observed as seen by the white oil curve in the same figure 5-F. The reaction film formation mechanism has been confirmed by microscopic examination of the wear scar surfaces to be discussed later. It seems that wear resistant conducting surface films have been formed which result in decreased wear and wear rates of En31 steel balls.

5.3 MICROSCOPIC EXAMINATION OF WEAR SCAR SURFACES AND RESULTS

The surfaces of all the wear scars after every wear test were examined with the help of Neophot Metallurgical Microscope. The results can be summarised as follows:-

(A) Dry Wear Test Results

(B) Lubricated Wear Test Results

Dry Wear Test Results:- The typical wear scars and appearance of the surface under the microscope are shown in figure 5-G. The increase in wear scar dimensions is seen as the sliding distance(or time) has been increased during a particular test.

Lubricated Wear Test Results:- The microscopic observations of wear scar surfaces of the lubricated wear test can be grouped into two categories as follows:-

(1) Tests using white oil as lubricant.

(2) Tests using elemental sulfur as additive and white oil as base oil.

(1) Tests Using White Oil as Lubricant:- The typical wear scars and appearance of the scar surfaces under the microscope are shown in figure 5-H. The increase in wear scar dimensions is seen as the sliding distance (or

time) has been increased during a particular test.

(2) <u>Tests Using Elemental Sulfur As Additive and White Oil as Base Oil</u>

To study the effect of concentration of sulfur in white oil on the characteristics of wear scar surfaces, the wear scars corresponding to step wear tests and continucus wear tests of the lubricated wear test series were examined in two ways:-

* Firstly the wear scars were examined immediately after the wear test in the as-such state.
* Secondly, a standard etching treatment was given by static immersion of the worn steel ball in 4% Nital for exactly one minute followed by microscopic examination of the etched surface of the wear scar.

<u>White Oil Containing0.1% sulfur</u>:- Figure 5-I shows photographs of the final wear scar after 3 step wear tests of 30 minutes duration each using white oil with 0.1% S as lubricant. This scar represents low sulfur higher wear regime. The surface shows severe wear with deep scratches and scratch pattern indicates sliding direction. There is no continuous film visible in the scar. The same wear scar, after standard etching treatment, has been shown in photograph 3 of figure 5-I. The wear scar surface has been almost entirely etched away.

this wear scar.

The microscopic investigation of wear scar surfaces indicate that the antiwear action of elemental sulfur in white oil is due to the wear [illegible] resistant barrier films formed in the antiwear domain. These barrier films are probably responsible for antiwear action. The possibility of such barrier films that can promote antiwear action has been discussed by Sethuramiah et.al.[26]. The barrier films can influence the rate of reaction of sulfur.

CHAPTER - 6 CONCLUSIONS

The wear volume Vs time behaviour of En 31 steel for dry wear condition is linear within initial stages.

With white oil, a slow wear rate is initially observed which is followed by a rapid transition to high wear which may be probably due to surface fatigue effect, and at still longer times, the wear volume levels off due to hydrodynamic film formation.

The lubricated wear volume Vs time behaviour of the white oil changes to a different mode on the addition of elemental sulfur as an additive into the white oil. A high wear rate is initially observed and the wear volume Vs time relationship shows gradual decrease in wear rate with increase in time. The initial wear rate as well as the subsequent decrease in wear rate with time, both seem to depend upon sulfur level in the lubricant. The initial wear rate is found to be minimum with 0.5% sulfur as compared with 0.1% and1% sulfur levels. The decrease in wear rate with time is also found to be maximum with 0.5% sulfur as compared with 0.1% and 1% sulfur levels. The step wear tests done show similar behaviour.

In dry wear tests, there is an initial linear regime for the Wear Volume Vs Metallic Contact relationship.

This indicates that wear volume and metallic contact can be quantitatively related. The linear relationship tends to be non-linear at longer running times which has been attributed to the formation of oxide films.

In lubricated wear tests, the relationship between wear volume Vs Metallic contact is found to be linear in the case of white oil alone as lubricant though there was increasing hydrodynamic effects with running time.

Addition of sulfur modifies the wear volume Vs metallic contact relationships. In this case the relationships are linear initially and become non-linear beyond a certain running time. The slope of the linear portion depends on sulfur concentration and shows a minimum value at 0.5% concentration. The non-linearity is attributed to changes that occur in the films with running time. Maximum levelling off is found in the case of 0.5% sulfur and is attributed to the formation of wear resistant areas in the friction surface. This has been later confirmed by miscroscopic observations.

Miscroscopic examination of the wear scars with low sulfur-high wear regime (0.1% sulfur) indicate inadequate surface film formation which enhances wear. Observation of

the surfaces with optimum sulfur concentration of 0.5% shows the surfaces are covered with reaction films and wear resistant nonetching zones. Observations of the surfaces in the high sulfur i.e. 2% sulfur corresponding to high sulfur high wear regime indicate that the surface film flakes off giving rise to high wear.

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APPENDIX - I INITIAL CONTACT STRESS IN THE PRESENT LUBRICANT TESTER

The Hertz theory of elastic deformation is used as a basis for calculating the initial contact area and contact stress[27] that exist between the fixed En 31 steel ball loaded on the moving En 31 steel disc under a normal load of 4 Kgs.

Referring figure 6-A, there exist two stages, the stage (A) in which two spheres are in contact, and stage (B) in which a sphere is in contact with a flat surface.

Consider stage (A), the formula for radius of contact a, for two spheres in contact is given below:-

$$a = \left[\frac{(3/4)\pi P(K_1 + K_2) R_1 R_2}{(R_1 + R_2)} \right]^{1/3}$$

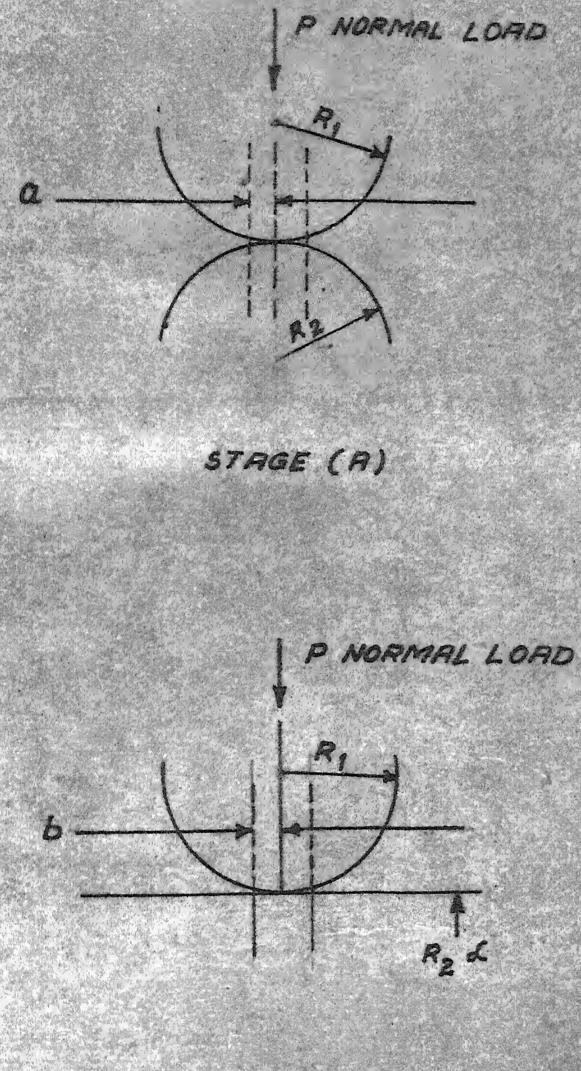
where P = Applied load, lb

$K_1 = K_2$ = elastic constants for the metals used (En 31 steel pair in this case)

$$K_1 = K_2 = \frac{1 - V^2}{E}$$

V = Poisson' ratio = 0.3

E = Young's Modulas of steel = 30×10^6 psi



STAGE (B)

FIG. - 6-A

$$K_1 = K_2 = \frac{1 - (0.3)^2}{30 \times 10^6} = 3.03 \times 10^{-8}$$

$$K_1 + K_2 = 0.606 \times 10^{-7} \ \frac{in^2}{Lb}$$

R_1, R_2 = Radii of curvature in inches

Considering stage (B), the formula for radius of contact is given for the case of sphere and a plane, as follows:-

$$R_2 = \infty$$

$$R = \left[(3/4)\pi P (K_1 + K_2) R_1 \right]^{1/3}$$

<u>Calculations</u>:-

P Normal load = 4 Kg = 8.82 lbs.

$$K_1 + K_2 = 0.606 \times 10^{-7} \ \frac{inch^2}{lb}$$

R_1 = 6.35 mm = 0.25 inches

R_2 = 40 mm = 1.5748 inches

$R_1 R_2 = 0.3937 \ inch^2$

$R_1 + R_2$ = 1.825 inches

$$a = \left(\frac{3}{4} \times \pi \times \frac{8.82 \times 0.606 \times 10^{-7} \times 0.3937}{1.825} \right)^{1/3}$$

$$a = \left(2.716 \times 10^{-7}\right)^{\frac{1}{3}}$$

$$= 6.459 \times 10^{-3} \text{ inches}$$

Similarly,

$$b = \left(\frac{3}{4} \times \pi \times 8.82 \times 0.606 \times 10^{-7} \times 0.25\right)^{\frac{1}{3}}$$

$$= \left(3.148 \times 10^{-7}\right)^{\frac{1}{3}} = 6.789 \times 10^{-3} \text{ inches}$$

$$\text{Stress} = \frac{\text{Load}}{\text{Area}}$$

$$\text{Area} = \pi . a . b = \pi \times 6.459 \times 10^{-3} \times 6.789 \times 10^{-3}$$

$$= 137.759 \times 10^{-6} \text{ inch}^2$$

$$\text{Stress} = \frac{8.82}{137.759 \times 10^{-6}}$$

$$= 6.4029 \times 10^{4} \text{ psi}$$

It is seen that the maximum pressure between the disc and the ball in present experimental conditions is below the elastic limit of the steel.

APPENDIX – II WEAR VOLUME DETERMINATION [28]

A wear scar is produced on the steel ball due to rubbing against the moving steel disc under the concentrated load. Due to the radius of curvature of the moving disc, the wear scar is not having a plane base but it is rather depressed inwards as shown in figure 6-B. Therefore the wear volume will be a function of the steel ball radius as well as the steel disc radius, assuming that the depressed zone of the worn ball takes the radius of curvature of the steel disc.

The total wear volume corresponding to a particular scar radius is equal to the volume of a convex lens as shown in the figure 6-C where one side radius of curvature is that of the steel ball and the opposite side radius of curvature is that of the steel disc.

Volume of the segment with one base can be calculated for each half of the Wear Volume Zone represented by the lens.

"Ball Wear Volume" corresponds to wear volume of the segment due to steel ball with one base as the wear scar circle.

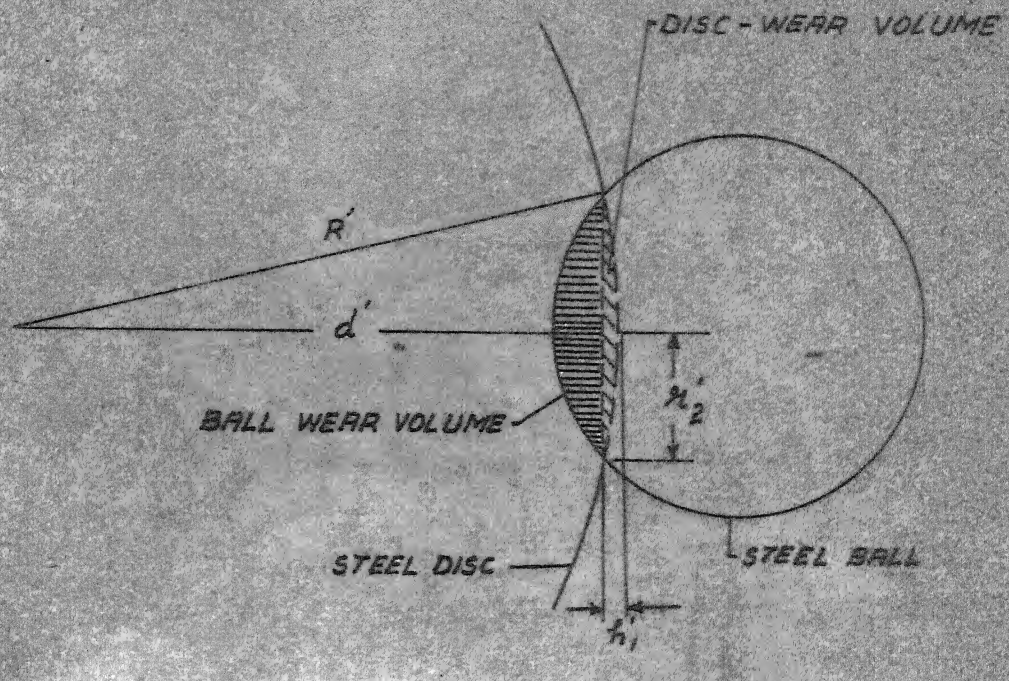

DISC - WEAR VOLUME
R'
d'
r_2'
BALL WEAR VOLUME
STEEL DISC
STEEL BALL
h_1'

FIG.- 6-C

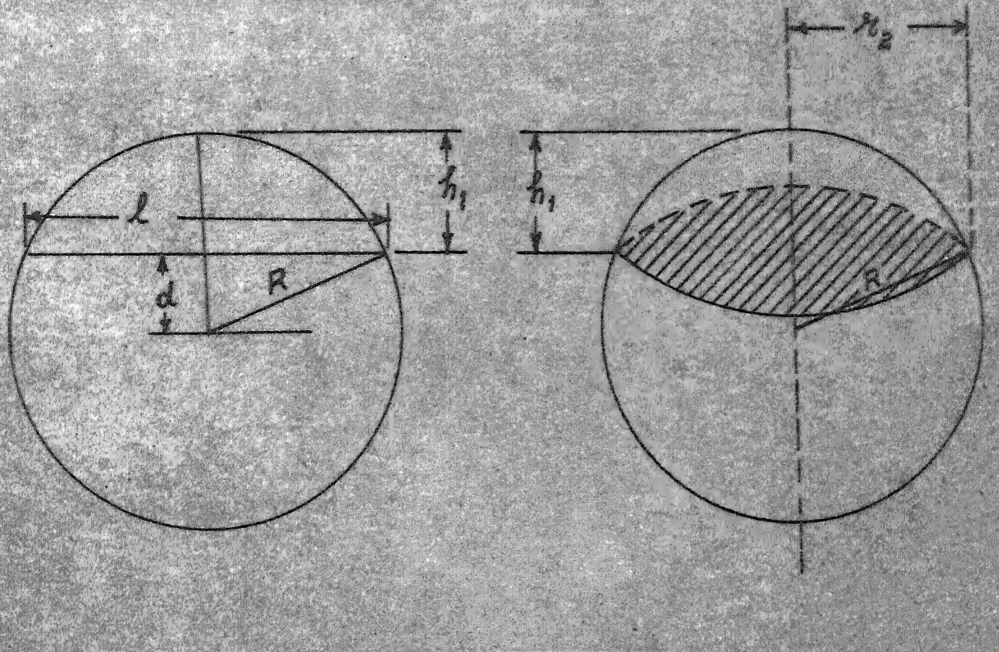

r_2
l
h_1
h_1
R
d
R

FIG.- 6-D

"Disc Wear Volume" corresponds to wear volume of the segment in the inward depressed zone of the wear scar due to the radius of curvature of the moving steel disc.

Total wear volume obviously is the sum total of the ball wear volume and disc wear volume for given value of the scar diameter.

Calculations for "Ball Wear Volume"

Referring figure 6-D, volume of spherical segment with one base is

$$V = \frac{\pi h_1}{6} \left(3r_2^2 + h_1^2 \right)$$

Refer figure 6-D,

$$h_1 = R - d = R - \sqrt{R^2 - (L/2)^2}$$

$$= R\left\{1 - \tfrac{1}{2}\sqrt{4 - (L/R)^2}\right\}$$

R = Ball radius = 6.35 mm

L = Wear Scar Diameter in mm

Hence, Wear Volume is given by,

$$B_{WV} = \frac{\pi}{6} R \left\{ 1 - \tfrac{1}{2}\sqrt{4 - (L/R)^2} \right\} \quad x$$

$$\left[3\left(\frac{L}{2}\right)^2 + R^2 \left\{ 1 - \tfrac{1}{2}\sqrt{4 - (L/R)^2} \right\}^2 \right]$$

Calculations for "Disc Wear Volume"

As shown in figure 6-C, volume of the spherical segment corresponding to the wear due to the curvature of the disc is,

$$V = \frac{\pi h_1'}{6}\left(3r_2^{1\,2} + h_1'^{\,2}\right)$$

But $r_2^1 = r_2$ (Calculations correspond to the same wear scar as made by the steel ball and steel disc rubbing together)

And $h_1^1 = R^1 - d^1$, R^1 = Disc's Radius of curvature

$2r_2^1 = 2r_2 = L$ = wear Scar Diameter

Hence,

Disc Wear Volume is given by,

$$D_{WV} = \frac{\pi}{6} R^1\left\{1-1/2\sqrt{4 - (L/R^1)^2}\right\}\left[3\left(\frac{L}{2}\right)^2 + R'^2\left\{1-\frac{1}{2}\sqrt{4-\left(\frac{L}{R'}\right)^2}\right\}^2\right]$$

Calculation for Total Wear Volume

Giving various values from0.4 mm to 3.0 mms as wear scar diamer, Bal Wear Volumes and corresponding Disc Wear volumes were determined and tabulated. The total wear volume was obtained by adding the Ball Wear Volume with its corresponding Disc Wear Volume for each value of the wear scar diameter.

A master curve is then prepared between Wear Volume and Scar diameter. The required values of Wear Volumes were obtained from this master curve. Typical Calculation Tables and master curve are shown in the following pages.

BALL-WEAR VOLUME $V_{BAL} = \frac{\pi}{6} R \left\{1-\frac{1}{2}\sqrt{4-(\ell/R)^2}\right\}\left[3\left(\frac{\ell}{2}\right)^2 + R^2\left\{1-\frac{1}{2}\sqrt{4-(\ell/R)^2}\right\}^2\right] = \frac{\pi}{6}\cdot\frac{3\ell^2}{4}.F + \frac{\pi}{6} F^3$ WHERE $F = R - \frac{R}{2}\sqrt{4-(\ell/R)^2}$

$= R - \frac{1}{2}\sqrt{4R^2 - \ell^2}$

AV. SCAR DIA. $-\ell-$ mm.	BALL DIAMETER $-2R-$ mm	$R = 6.35$ $2R+\ell$	$2R-\ell$	$\frac{1}{2}\sqrt{4R^2-\ell^2}$	$F = R-\frac{1}{2}\sqrt{4R^2-\ell^2}$	$\frac{\pi\ell^2}{8}\cdot F$	$\frac{\pi}{6} F^3$
·9729	12·7	13·6729	11·7271	6·331340015	·018659985	6.9359×10^{-3}	3.39×10^{-6}
·9568	12·7	13·6568	11·7432	6·331953365	·018046635	6.4878×10^{-3}	3.07×10^{-6}
·9959	12·7	13·6959	11·7041	6·33044594	·01955406	7.61602×10^{-3}	3.91×10^{-6}
·92	12·7	13·62	11·78	6·333316666	·01668333	5.5452×10^{-3}	2.43×10^{-6}
·9315	12·7	13·6315	11·7685	6·33289641	·01710359	5.828×10^{-3}	2.62×10^{-6}
1·0051	12·7	13·7051	11·6949	6·33008242	·01991758	7.901599×10^{-3}	4.14×10^{-6}
1·1247	12·7	13·8247	11·5753	6·32505039	·02494961	1.2394×10^{-2}	8.1319×10^{-6}

CALCULATION TABLE NO. C1-B

Disc Wear Volume

$$V_{DISC} = \frac{\pi}{6} R'\left\{1-\tfrac{1}{2}\sqrt{4-(\ell/R')^2}\right\}\left[3\left(\ell/2\right)^2 + R'^2\left\{1-\tfrac{1}{2}\sqrt{4-(\ell/R')^2}\right\}^2\right] = \frac{\pi}{6}\frac{3\ell^2}{4}F' + \frac{\pi}{6}F'^3$$

where $F' = R' - R'/2\sqrt{4R'^2 - \ell^2}$

$$= R'\left\{1-\tfrac{1}{2}\sqrt{4-(\ell/R')^2}\right\}$$

ℓ Av. Scar dia mm	Disc dia $2R'$ mm	Disc radius R' mm	$2R'+\ell$	$2R'-\ell$	$\frac{1}{2}\sqrt{4R'^2-\ell^2}$	$F' = R'-\frac{1}{2}\sqrt{4R'^2-\ell^2}$	$\frac{\pi\ell^2}{8}\cdot F'$	$\frac{\pi}{6}\cdot F'^3$
·9729	40	20	40·9729	39·0271	19·9940833	·005917	2.199×10^{-3}	1.085×10^{-7}
·9568	40	20	40·9568	39·0432	19·9942775	·0057224	2.057×10^{-3}	9.812×10^{-8}
·9959	40	20	40·9959	39·0041	19·9938002	·00619982	2.4147×10^{-3}	1.247×10^{-7}
·92	40	20	40·92	39·08	19·994709	·0052907	1.759×10^{-3}	7.75×10^{-8}
·9315	40	20	40·9315	39·0685	19·9945761	·0054238	1.848×10^{-3}	8.35×10^{-8}
1·0051	40	20	41·0051	38·9949	19·9936851	·00631491	2.51×10^{-3}	1.32×10^{-7}
1·1247	40	20	41·1247	38·8753	19·9920925	·00790751	3.93×10^{-3}	2.59×10^{-7}

CALCULATION TABLE NO. C2-A

Ball-Wear Volume

AV. SCAR DIAMETER mm	Ball Diameter mm	Ball Radius mm	$2R+\ell$	$2R-\ell$	$\frac{1}{2}\sqrt{4R^2-\ell^2}$	$F = R-\frac{1}{2}\sqrt{4R^2-\ell^2}$	$\frac{\pi\ell^2}{8}\cdot F$	$\frac{\pi}{6}\cdot F^3$
1.2	12.7	6.35	13.9	11.5	6.32158998	.028410015	1.6065×10^{-2}	1.2×10^{-5}
1.4	12.7	6.35	14.1	11.3	6.31129939	.03870061	2.9787×10^{-2}	3.03×10^{-5}
1.6	12.7	6.35	14.3	11.1	6.29940473	.05059527	5.0864×10^{-2}	6.78×10^{-5}
1.8	12.7	6.35	14.5	10.9	6.28589691	.06410309	8.156×10^{-2}	1.379×10^{-4}
2.0	12.7	6.35	14.7	10.7	6.2707655	.0792345	1.2446×10^{-1}	2.604×10^{-4}
2.2	12.7	6.35	14.9	10.5	6.25399872	.09600128	1.825×10^{-1}	4.63×10^{-4}
2.4	12.7	6.35	15.1	10.3	6.23558337	0.11441663	2.588×10^{-1}	7.84×10^{-4}
2.6	12.7	6.35	15.3	10.1	6.215504805	0.134495195	3.57×10^{-1}	1.274×10^{-3}
2.8	12.7	6.35	15.5	9.9	6.193746845	0.156253155	4.81×10^{-1}	1.998×10^{-3}
3.0	12.7	6.35	15.7	9.7	6.170291725	0.17970828	6.35×10^{-1}	3.04×10^{-3}
1.5	12.7	6.35	14.2	11.2	6.30555311	0.044446895	3.927×10^{-2}	4.598×10^{-5}
2.5	12.7	6.35	15.2	10.2	6.22575297	0.12424704	3.05×10^{-1}	1.004×10^{-3}

DISC-WEAR VOLUME

ℓ AV. SCAR DIAMETER mm	$2R'$ DISC DIAMETER mm	R' DISC RADIUS mm	$2R'+\ell$	$2R'-\ell$	$\frac{1}{2}\sqrt{4R'^2-\ell^2}$	$F' = R'-\frac{1}{2}\sqrt{4R'^2-\ell^2}$	$\frac{\pi\ell^2}{8}\cdot F'$	$\frac{\pi}{6}\cdot F'^3$
1.2	40	20	41.2	38.8	19.99099797	.00900203	5.09×10^{-3}	3.82×10^{-7}
1.4	40	20	41.4	38.6	19.9877462	.01225376	9.43×10^{-3}	9.63×10^{-7}
1.5	40	20	41.5	38.5	19.9859325	.01406745	1.243×10^{-2}	1.46×10^{-6}
1.6	40	20	41.6	38.4	19.9839936	.01600641	1.61×10^{-2}	2.15×10^{-6}
1.8	40	20	41.8	38.2	19.9797397	.0202602	2.578×10^{-2}	4.35×10^{-6}
2.0	40	20	42.0	38.0	19.9749844	.02501565	3.93×10^{-2}	8.196×10^{-6}
2.2	40	20	42.2	37.8	19.9697271	.030273	5.75×10^{-2}	1.45×10^{-5}
2.4	40	20	42.4	37.6	19.9639675	.03603246	8.15×10^{-2}	2.45×10^{-5}
2.5	40	20	42.5	37.5	19.9608993	.03910073	9.596×10^{-2}	3.13×10^{-5}
2.6	40	20	42.6	37.4	19.957705	.0422947	1.1227×10^{-1}	3.96×10^{-5}
2.8	40	20	42.8	37.2	19.95093982	.04906018	1.510×10^{-1}	6.18×10^{-5}
3.0	40	20	43	37	19.9436707	.05632933	1.991×10^{-1}	9.36×10^{-5}

CALCULATION TABLE NO. C3

AV. SCAR DIA. mm	BALL WEAR VOLUME mm^3	DISC WEAR VOLUME mm^3	TOTAL WEAR VOLUME mm^3
0.4255	2.53464×10^{-4}	8.04549×10^{-5}	3.339189×10^{-4}
0.4347	2.76120×10^{-4}	8.7641×10^{-5}	3.63761×10^{-4}
0.4531	3.259199×10^{-4}	1.03450×10^{-4}	4.293699×10^{-4}
0.4922	4.5379×10^{-4}	1.4405×10^{-4}	5.9784×10^{-4}
0.41055	2.1967×10^{-4}	6.974×10^{-5}	2.8941×10^{-4}
0.50945	5.20927×10^{-4}	1.653349×10^{-4}	6.862619×10^{-4}
0.5175	5.5423×10^{-4}	1.7603×10^{-4}	7.3026×10^{-4}
0.53935	6.5444×10^{-4}	2.07703×10^{-4}	8.62143×10^{-4}
0.54855	7.00267×10^{-4}	2.2224×10^{-4}	9.22507×10^{-4}
0.5589	7.5464×10^{-4}	2.3949×10^{-4}	9.9413×10^{-4}
0.6003	1.00441×10^{-3}	2.441559×10^{-4}	1.248566×10^{-3}
0.6325	1.2379×10^{-3}	3.9283×10^{-4}	1.63073×10^{-3}
0.6509	1.3883×10^{-3}	4.4057×10^{-4}	1.82887×10^{-3}
0.6762	1.61729×10^{-3}	5.13175×10^{-4}	2.130465×10^{-3}
0.6923	1.7766×10^{-3}	5.6381×10^{-4}	2.34041×10^{-3}
0.67735	1.62838×10^{-3}	5.1668×10^{-4}	2.14506×10^{-3}

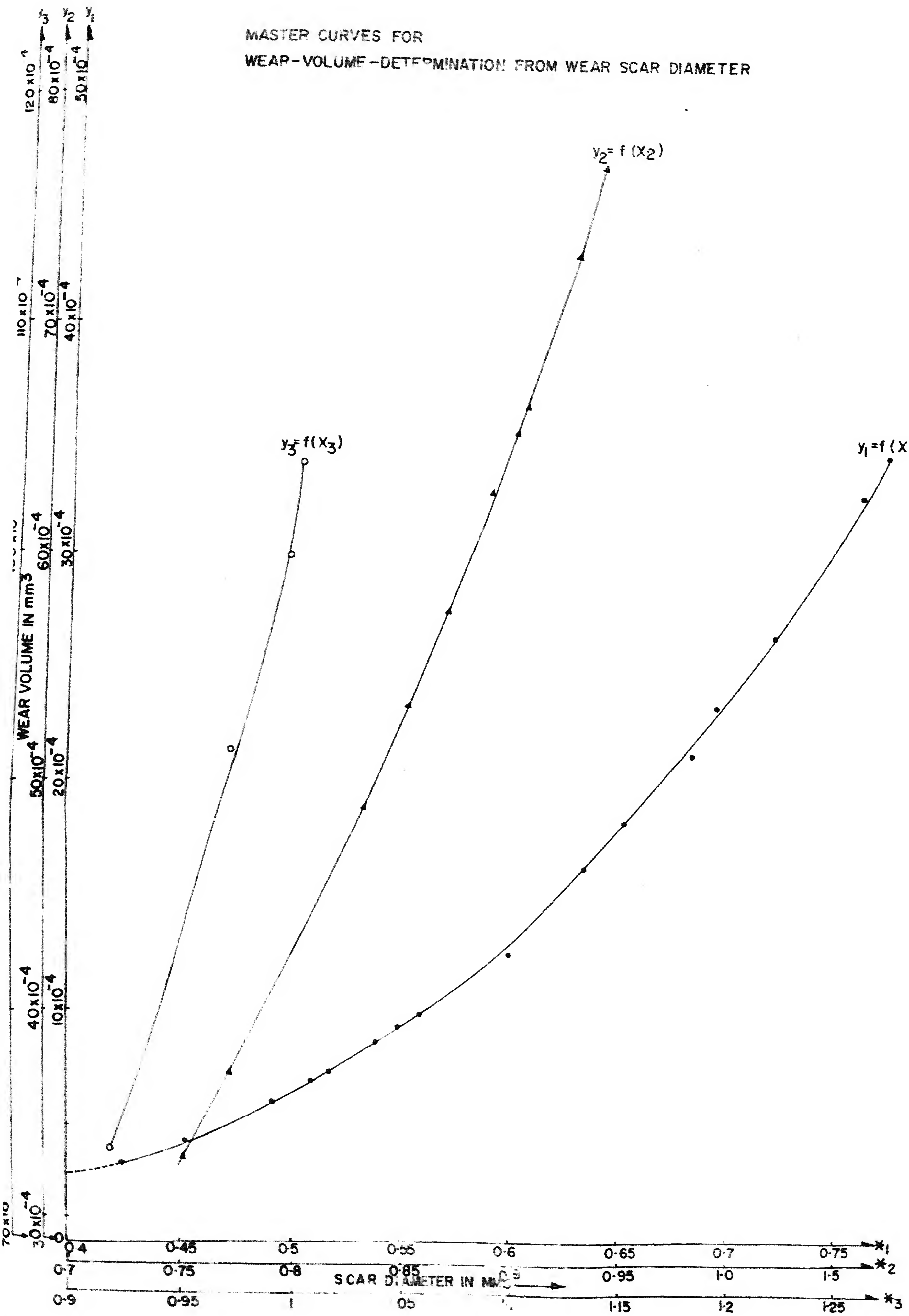
CALCULATION TABLE NO. C4

AV. SCAR DIA. mm	BAL WEAR VOLUME mm^3	DISC WEAR VOLUME mm^3	TOTAL WEAR VOLUME mm^3
0.71415	2.0123×10^{-3}	6.384×10^{-4}	2.6507×10^{-3}
0.7521	2.4754×10^{-3}	7.853×10^{-4}	3.2607×10^{-3}
0.76245	2.6147×10^{-3}	8.295×10^{-4}	3.4442×10^{-3}
0.7728	2.7597×10^{-3}	8.7548×10^{-4}	3.63518×10^{-3}
0.8326	3.7188×10^{-3}	1.17959×10^{-3}	4.89839×10^{-3}
0.851	4.05842×10^{-3}	1.287×10^{-3}	5.3454×10^{-3}
0.8671	4.3819×10^{-3}	1.3876×10^{-3}	5.7695×10^{-3}
0.8855	4.7644×10^{-3}	1.509×10^{-3}	6.2734×10^{-3}
0.8947	4.9595×10^{-3}	1.5729×10^{-3}	6.5324×10^{-3}
0.8993	5.06244×10^{-3}	1.5898×10^{-3}	6.65224×10^{-3}
0.92	5.5452×10^{-3}	1.759×10^{-3}	7.3042×10^{-3}
0.9315	5.828×10^{-3}	1.848×10^{-3}	7.676×10^{-3}
0.9568	6.4878×10^{-3}	2.057×10^{-3}	8.5448×10^{-3}
0.9729	6.9359×10^{-3}	2.199×10^{-3}	9.1349×10^{-3}
0.9959	7.6160×10^{-3}	2.4147×10^{-3}	1.00307×10^{-2}
1.0051	7.9015×10^{-3}	2.51×10^{-3}	1.04115×10^{-2}
1.1247	1.2394×10^{-2}	3.93×10^{-3}	1.6324×10^{-2}

CALCULATION TABLE NO. C5

AVERAGE SCAR DIA. mm	BALL WEAR VOLUME mm^3	DISC WEAR VOLUME mm^3	TOTAL WEAR VOLUME mm^3
1.2	$1.6.65 \times 10^{-2}$	5.09×10^{-3}	2.1155×10^{-2}
1.4	2.9787×10^{-2}	9.43×10^{-3}	3.9217×10^{-2}
1.5	3.927×10^{-2}	1.243×10^{-2}	5.17×10^{-2}
1.6	5.0864×10^{-2}	1.61×10^{-2}	6.6964×10^{-2}
1.8	8.156×10^{-2}	2.578×10^{-2}	1.0734×10^{-1}
2.0	1.2446×10^{-1}	3.93×10^{-2}	1.6376×10^{-1}
2.2	1.825×10^{-1}	5.75×10^{-2}	2.4×10^{-1}
2.4	2.588×10^{-1}	8.15×10^{-2}	3.403×10^{-1}
2.5	3.05×10^{-1}	9.596×10^{-2}	4.0096×10^{-1}
2.6	3.57×10^{-1}	1.1227×10^{-1}	4.6927×10^{-1}
2.8	4.81×10^{-1}	1.510×10^{-1}	6.32×10^{-1}
3.0	6.35×10^{-1}	1.991×10^{-1}	8.341×10^{-1}

MASTER CURVES FOR
WEAR-VOLUME-DETERMINATION FROM WEAR SCAR DIAMETER



APPENDIX - III

BASIC OBSERVATIONS OF WEAR SCARS' DIMENSIONS IN VARIOUS WEAR TESTS

Expt.No.	Wear Scar Measurements (Divisions) Major dia.	Minor dia.	Av.Dia.	Factor of Multiplication	Wear Scar Av. dia in mms.
(1)	(2)	(3)	(4)	(5)	(6)
1 DCm_1	380	351	366	1 div.=.0023 mm	0.8418
1 DCm_2	527	435	481	-do-	1.1063
1 DCm_3	577	527	552	-do-	1.2696
1 DCm_4	643	562	603	-do-	1.3869
2-DCm_1	506	452	479	-do-	1.1017
2-DCm_2	622	553	588	-do-	1.3524
3-DCm_1	688	581	635	-do-	1.4605
3-DCm_2	72	62	67	1 mm = 40 divisions	1.675
4-DCm_1	606	524	563	1 mm = 442 divisions	1.2949
4-DCm_2	650	540	595	-do-	1.3685
4-DCm_3	679	551	615	-do-	1.4145
5-DCm_1	665	508	584	-do-	1.3432
5-DCm_2	60	51	56	1 mm = 40 divisions	1.400
5-DCm_3	65	54	60	-do-	1.500
6-DCm_1	460	400	430	1 mm = 442 divisions	0.989
6-DCm_2	615	570	593	-do-	1.3639

(1)	(2)	(3)	(4)	(5)	(6)
$6\text{-}DCm_3$	700	600	650	1 mm = 442 divisions	1.495
$6\text{-}DCm_4$	770	690	730	-do-	1.679
$6\text{-}DCm_5$	812	726	769	-do-	1.7687
$7\text{-}DCm_1$	685	592	639	-do-	1.4697
$7\text{-}DCm_2$	839	709	774	-do-	1.7802
$7\text{-}DCm_3$	82	74	78	1 mm = 40 divisions	1.950
$8\text{-}DCm_1$	1.95 mm	1.62 mm	1.785	-direct-	1.785
$8\text{-}DCm_2$	2.2 mm	1.9 mm	2.05	-direct-	2.05
$8\text{-}DCm_3$	2.30 mm	2.00 mm	2.15	-direct-	2.15
$9\text{-}DCm_1$	94	78	86	1 mm = 40 divisions	2.15
$9\text{-}DCm_2$	2.75	2.65	2.700	-direct-	2.70
$9\text{-}DCm_3$	2.8	2.7	2.750	-direct-	2.75
$10\text{-}DCm_1$	2.25	1.90	2.075	-direct-	2.075
$10\text{-}DCm_2$	2.3	2.05	2.175	-direct-	2.175
$1\text{-}LSm_1$	386	309	348	1 mm = 442 Divisions	0.8004
$1\text{-}LSm_2$	410	338	374	-do-	0.8602
$1\text{-}LSm_3$	441	360	401	-do-	0.9223
$2\text{-}LSm_1$	500	406	453	-do-	1.0419

(1)	(2)	(3)	(4)	(5)	(6)
2-LSm_2	509	428	469	1 mm = 442 Divisions	1.0787
2-LSm_3	538	439	489	-do-	1.1247
3-LSm_1	432	345	389	-do-	0.8947
3-LSm_2	437	349	393	-do-	0.9039
3-LSm_3	440	353	397	-do-	0.9131
4-LSm_1	360	290	325	-do-	0.7475
4-LSm_2	362	292	327	-do-	0.7521
4-LSm_3	362	292	327	-do-	0.7521
5-LSm_1	332	263	298	-do-	0.6854
5-LSm_2	335	264	300	-do-	0.6900
5-LSm_3	335	269	302	-do-	0.6946
6-LSm_1	351	278	315	-do-	0.7245
6-LSm_2	351	278	315	-do-	0.7245
6-LSm_3	355	280	318	-do-	0.7314
7-LSm_1	368	308	338	-do-	0.7774
7-LSm_2	380	318	349	-do-	0.8027
7-LSm_3	389	320	355	-do-	0.8165
8-LSm_1	454	377	416	-do-	0.9568
8-LSm_2	475	391	433	-do-	0.9959

(1)	(2)	(3)	(4)	(5)	(6)
8-LSm_3	478	400	439	1 mm = 442 Divisions	1.0097
1-LCm_1	214	143	179	-do-	0.41055
1-LCm_2	245	180	213	-do-	0.48875
1-LCm_3	251	218	235	-do-	0.53935
1-LCm_4	259	218	238	-do-	0.5428
2-LCm_1	257	186	222	-do-	0.50945
2-LCm_2	275	211	243	-do-	0.5612
2-LCm_3	289	233	261	-do-	0.6003
3-LCm_1	310	240	375	-do-	0.6325
3-LCm_2	349	272	311	-do-	0.71415
3-LCm_3	369	303	336	-do-	0.7728
4-LCm_1	331	258	295	-do-	0.67735
4-LCm_2	363	300	332	-do-	0.76245
5-LCm_1	371	300	336	-do-	0.7728
5-LCm_2	414	339	377	-do-	0.8671
5-LCm_3	428	341	385	-do-	0.8855
1-LC	243	181	212	-do-	0.4876
2-LC	248	186	217	-do-	0.4991
3-LC	259	195	227	-so-	0.5221

(1)	(2)	(3)	(4)	(5)	(6)
4-LC	263	204	234	1 mm = 442 Divisions	0.53705
5-LC	264	191	228	-do-	0.5244
6-LC	274	203	239	-do-	0.54855
7-LC	293	221	257	-do-	0.5911
8-LC	300	235	268	-do-	0.61525
9-LC	310	244	277	-do-	0.6371
10-LC	331	263	297	-do-	0.6831
11-LC	364	280	322	-do-	0.7406
12-LC	364	293	329	-do-	0.7567
13-LC	381	314	348	-do-	0.8004
14-LC	400	314	357	-do-	0.8211
15-LC	408	303	356	-do-	0.81765
16-LC	239	193	216	-do-	0.4968
17-LC	238	197	218	-do-	0.5014
18-LC	234	271	253	-do-	0.5819
19-LC	314	267	291	-do-	0.6693
20-LC	339	281	310	-do-	0.7130
21-LC	371	313	342	-do-	0.7866
22-LC	375	319	347	-do-	0.7981
23-LC	390	329	360	-do-	0.8280
24-LC	410	359	385	-do-	0.8855
25-LC	435	359	397	-do-	0.9131
26-LC	504	413	459	-do-	1.0557

(1)	(2)	(3)	(4)	(5)	6
27-LC	542	447	495	1 mm = 442 Divisions	1.1385
28-LC	557	461	509	-do-	1.1707
29-LC	693	580	637	-do-	1.4651
30-LC	231	185	208	-do-	0.4784
31-LC	265	219	242	-do-	0.5566
32-LC	277	214	246	-do-	0.5658
33-LC	287	225	256	-do-	0.5888
34-LC	330	267	299	-do-	0.6877
35-LC	337	272	304	-do-	0.6992
36-LC	302	254	278	-do-	0.6394
37-LC	330	262	296	-do-	0.6808
38-LC	318	263	291	-do-	0.6693
39-LC	342	279	311	442 Divs. = 1 mm.	0.7153
40-LC	359	307	333	-do-	0.7659
41-LC	390	333	362	-do-	0.8326
42-LC	209	170	190	-do-	0.437
43-LC	239	199	219	-do-	0.5037
44-LC	263	222	243	-do-	0.5589
45-LC	295	242	269	-do-	0.6187
46-LC	309	261	285	-do-	0.6555
47-LC	350	281	316	-do-	0.7268

(1)	(2)	(3)	(4)	(5)	(6)
48-LC	354	285	320	442 Divs. = 1 mm	0.7360
49-LC	355	295	325	-do-	0.7475
50-LC	425	355	390	-do-	0.8970
51-LC	509	420	465	-do-	1.0695
52-LC	473	404	438	-do-	1.0074
53-LC	557	459	508	-do-	1.1684
54-LC	482	391	437	-do-	1.0051
55-LC	453	380	417	-do-	0.9591
56-LC	461	390	426	-do-	0.9798